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SHOCK AND VIBRATION BULLETIN

MAY 1953

RESEARCH AND DEVELOPMENT BOARD

Department of Defense · Washington, D. C.

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The 20th Shock and Vibration Symposium was held at the Naval Research Laboratory, Washington 25, D. C., on May 12 - 13, 1953. The Navy was the host.

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Att: Mr. H. L. Solomonson 1

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Pres., Hqs., Army Field Forces,
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BuAer
Att: Code Aer-EL-57 5
Att: Code Aer-DE-223 5
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Att: Code SI-521 2
Att: Aer-SI-511 1
Att: Code Aer-AE-312 1
Att: Code AE-5 1
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Att: Code RE 10
Att: Code Re9 2
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Att: Code 371 2
Att: Code 436 2
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Att: Maj. Gen. F. A. Heileman 1

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Att: CDR. K. S. Brown 3
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Att: Weapons & Missiles Branch 1

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Att: Mr. H. W. Maas 2

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Att: Fire Control Lab. 1
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Att: ORDEV 1

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Att: Electronics Officer 1
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Att: Quality Evaluation Lab. 1
Att: K. K. Maki 1
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Att: Mr. V. S. Kupelian 2

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Johnsville
Att: Mr. L. P. Clark 1
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Att: Materials & Components Br. 1

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Att: Mr. C. P. Forester 1

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CDR, NAMC, NAF 1

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Att: Dr. R. Weller 1
Att: Library 1

Dir., Naval Air Test Center, Patuxent River
Att: Electronics Test Div. 1

CO, USNAS, Alameda
Att: Aeronautical Engr. Group 1

U. S. Naval Attache, Navy 100
Att: Logistics Div. 1

Supt., NGF
Att: Aviation Ordnance Unit 6
Att: Design Dept., Code DE-740 4
Att: Engineering Dept., Code 733.2 1

CDR, NOL, White Oak
Att: Technical Director 4
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Att: Code UW1 2
Att: Mr. J. W. Talcott 1
Att: Mr. D. E. Marlowe 1
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Med. OinC, USNMRI, Bethesda
Att: LCDR D. E. Goldman 1

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Att: Mr. P. W. Ott 1
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Att: Library 1

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Att: Technical Library 1

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Att: P80 1
Att: P8033 3

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Att: CDR. W. C. Hushing 1

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Att: Ship Design Section 1

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Att: Supply Engineering Div. 1

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Att: Dr. G. F. W. Mulders 1

ONR, Washington
Code 438 8
Code 346 1

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Patrick AFB
Att: Mr. A. H. Warner 1
Att: Chief, Tech. Systems Lab. 1

Northrop Aircraft Inc., Patrick AFB
Att: Mr. W. H. Broughton 1

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Navy 128
Att: Code 264 1

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Att: Mr. J. A. Vichness 1

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Ft. Trumbull
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Att: Mr. J. M. Taylor 1

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Att: Chief, Code WCRT 1
Att: Chief, Code WCLC 1
Att: Chief, Code WCLE 1
Att: Chief, Code WCRD 1
Att: Chief, Code WCLS 1
Att: Chief, Code WCLB 1
Att: Chief, Code WCLP 1
Att: Chief, Code WCLF 1
Att: Chief, Code WCRR 1
Att: Chief, Code WCL 1
Att: Chief, Code WCLO 1
Att: Chief, Code WCLG 1
Att: Dr. M. G. Scherberg 1
Att: Lt. R. B. Orensteen 1
Att: Mr. B. K. Wernicke 1
Att: Lt. Col. J. P. Stapp 1
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(via WCRTG) 1

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Att: Dr. R. Beeuwkes, Jr. 3

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CO, Picatinny Arsenal
Att: Col. C. R. Dutton 1

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Att: Code 251b 1
Att: Mr. E. C. Taylor 1

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Att: Design Supt., Code 251 1

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Att: Chief, Container Research Project 1

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Att: Military Planning Division 1

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Att: Technical Library 4
Att: Mr. H. C. Bolon 1

RDB Comm. on Electronics, New York
Att: Mr. Melvin Barmat 1

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CDR, San Francisco Naval Shipyard
Att: Code 250 1
Att: Code 254T 1

RDB
Att: Library 1
Att: Comm. on Electronics 1
Att: Comm. on Ordnance 1
Att: Comm. on Guided Missiles 1
Att: Comm. on Equipment and Materials 1
Att: Comm. on Aeronautics 1
Att: Navy Secretary 1

CO, Savanna Ordnance Depot
Att: Ammunition Instructors School 1

CG, SCEL, Fort Monmouth
Att: Mr. N. E. Lee, CSL 5
Att: Mr. H. L. Stout, SSL 5
Att: ESL 5

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Sands Point
Att: Library Branch 1

Foreward

This Bulletin represents many diversified areas in the widespread field of shock and vibration. It contains the proceedings of the 20th Symposium which recorded the largest registration for attendance in the history of the Shock and Vibration Centralizing Activity. Scientists and engineers from all over this country (some other countries were also represented) came to discuss current and pertinent technical material of common interest to them. The sustained enthusiasm shown throughout the several sessions indicated on the part of the delegates an eagerness to learn from and contribute to the discussions. Whether it is the type of program or the timeliness of its subject-matter or some other reason which attracted so many working technicians to this meeting is not clear to this writer. Whatever the cause, the planners and arrangers of these symposia are gratified with the response and attendance. We sincerely hope that the benefits derived from the meeting and its proceedings printed in this Bulletin will contribute to our national preparedness.

In attempting to disseminate current technical information to a heterogeneous group representing the various agencies in the Department of Defense, we strive to satisfy the needs of the greatest number. The requests from the working-level engineer and designer for shock and vibration topics to be discussed are always honored by the Interservice Technical Group to the extent of programming a particular subject whenever feasible. All the topics which were presented and discussed in the five sessions of the 20th Symposium, including the supplementary papers on the program, were selected from over one hundred subject-titles submitted to this Activity as requiring discussion. The large number of favorable comments regarding the symposium as a whole attests to the fact that the majority of delegates benefited from the meeting; and that is our primary objective.

To be sure, there were some in attendance who considered a couple of the papers too "high-brow" and theoretical; while perhaps an equal number regarded several of the topics as too elementary to be discussed at a technical meeting. However, the general reaction indicates that steady progress is being made in disseminating needed and useful information in this field. If you find that the material in this Bulletin is not at the level you expected it to be and would like a particular subject explored along certain lines, please submit your ideas for the next symposium. The Group will do its best to incorporate your suggestions in the program of the 21st Symposium.

Elias Klein

In This Issue

Attendance

Aberdeen Proving Grounds

Apgar, J. W.
Cronin, R. F.
Elder, A. S.
Zimmer, G. A.

Aerojet-General

Boulton, V. R.

AFSWP

Reynolds, J. R.
Savker, D.
Traylor, M. E.

Aircraft Armaments

Hirsch, R. A.
Kahn, M. J.

AiResearch

Winslow, R. W.

U. S. Air Force

Lamb, Carl

Allegany Ballistics Lab.

Heider, R. C.
Shank, Earl

Applied Design Co.

Aikman, J. B.

Applied Physics Lab., JHU

Blevins, R. W.
Caywood, W. C.
Mains, R. M.
Tellman, H. A.

Arma Corp.

Atalese, V.
Cammarata, J.

Armour Research

Dickens, R. G.
Mintz, Fred

Army - G-4

Wensel, H. T.

Army Field Forces

Sloan, H. B.

Army Transportation Corps

Hamilton, R. M.

Soc. of Automotive Engineers

Staiger, D. L.

Barry Corp.

Cavanaugh, R. D.
Crede, C. E.
Markowitz, J.
Young, S. E.

Beech Aircraft

Pierpont, W. G.

Bell Aircraft

Conley, H. D.
Costello, R.
D'Ewart, B.
Dobony, M.
Fero, L.
Friedenberg, J.
Morgan, R.
Ryken, J.
Wheeler, J.
Wittman, J.
Yeager, R.

Bell Telephone Labs.

Blackmore, R. W.
Jones, E. H.
Scott, J. W.

Bendix - Eclipse-Pioneer

Russo, J. P.

Bendix - K. C. Div.

Arnold, Gene

Bendix Radio

Archer, W. D.
Best, J. H.
Burns, C. O.
Cservek, J.
Cumings, K.
Edel, D. H.
Geidt, J.
Glass, H. D.
Goldberg, V. R.
Hawkins, E. C.
Kauffman, D.
Pariseau, R.
Reed, Emory
Secura, R.
Wong, B.
Wroten, J. F., Jr.

Bendix - Scintilla-Magneto

Hogan, E. P.
Tyne, J. J.

Birma Mfg. Co.

Feine, E.
Welton, H. R.

Boeing

Eldridge, D. F.
Gebhardt, G. T.
Trotter, W. D.

Borg Warner

Velander, L.

Boston Naval Shipyard

Antkowiak, E. T.
Danberg, A.

Bristol Co.

Sleeper, L. M.

British Joint Services Mission

Hyde, N. E.
Starks, J. F.

Buckeye Steel Castings

Settles, J. C.

BuAer

Arkin, S. M.
Burch, R. L.
Calapristi, A. F.
Colb, H. A.
Ellis, F. E., Jr.
Evensen, R. N.
George, J.
Gunsolley, V.
Hunter, H. F.
Mustin, G. S.
Paul, F.
Sanborn, E. L.
Schuldenfrei, M.
Seigel, S. J.
Sidle, W. R.

BuOrd

Goldberg, M.
Jackson, C. H.
Marquardt, E. A.
Mastracci, M. L.
Moskovitz, A. I.
Stanton, O. J.

BuShips

Bell, B. E.
Berg, David
Blazek, S. M.
Brown, K. S.
Cooper, S. S.
Deyo, B.
Feldman, S.
Gareau, R.
Henning, E. S.
Levine, G.

BuShips (Cont'd)

Losee, L. K.
Mathias, H. A.
Norris, T. B.
Oliver, R. H.
Parker, E. J.
Pratt, D.
Rassman, E.
Rosenberg, F. T.
Willman, L. N.

Calif. Inst. of Tech., JPL

McDonald, R. R.

Canadian Signals R. & D. Estab.

Simmons, E. J.

Capehart-Farnsworth

Lewis, J. M.

Chance-Vought

Lang, A. L., Jr.

Chrysler

Hammer, J. S.
Isley, M. J.
Janeway, R. N.
Smith, H. R.
Vehko, V. J.

Civil Aeronautics Adm.

Husic, W. J.
Rosenbaum, R.
Schroers, R. J.

Cleveland Penumatic Tool

Hogan, W. H.

Coles Signal Lab.

Ryder, H. E., Jr.
Thorpe, C. L.
Webber, H. K.

Collins Radio

Dunns, R. S.
Toms, Ed

Columbia University

Bleich, H. H.

Conn. Hard Rubber

Mordarski, Walter
Pfenninger, Arnold

Consolidated Vultee

Brady, M. C.
Cheney, H. K.
Friedrich, H. R.
Morrow, W. J.

Container Labs.

Wharton, T. P.

Cook Electric

Gallagher, W. E.

Cornell Aeronautical Labs.

Andreopoulos, T.
Newman, M. K.

Curtiss-Wright

Pratt, W.

David Taylor Model Basin

Bort, R. L.
Carr, W. E.
Chertock, G.
Clark, E. S.
Elmer, G. D.
Kennard, E. H.
Kinsey, C. H.
Rich, H. L.
Roberson, L. W.
Ruggles, R. D.
Schutt, W. D.
Sette, W. J.
Vane, F. F.
Wagner, E. R.
Weinberger, F.
Wigle, B. M.

Defense Research Board

Dumas, J. E.

Detroit Arsenal

Brookes, J. G.
Davis, P. S.
Mazur, W. E.
Stroup, F. H.
VanCauwenberg, P. H.

Douglas Aircraft

Frank, C. W.
Karnesky, A. L.

Edo Corp.

Klurfeld, S.

EEC Naval Headquarters

Poushinsky, A. W.

Eglin, AFAC

Bostwick, L. C.
Greene, R. E.

Electric Boat

Conner, W. L.
McClure, A. C.

ERDL Fort Belvoir

Carr, K. M.
Engle, C. T.
Maas, H. W.
Madden, E. J.
McKay, R. L.
Pusey, H. C.
Stowell, F.

Engineering & Research Corp.

Dorr, G. W.

Evans Signal Lab.

Biamonte, O.
Devroetes, P.

Fairchild

Dolkas, C. B.
Ebersole, H. N.
Gregory, A. T.
Marsden, J. N.
Sellers, W. H.
Veterito, A. M.
Williams, E. L.

Farmingdale Corp.

Craig, R. W.
Fellows, W. E.
Focht, N. S.
Sherwood, W. A.

Firestone

Deist, H. H.

Forest Products

Jones, R. E.

Frankford Arsenal

Albert, Vernon
Askins, David

Frankford Arsenal (Cont'd)

DeWald, John
Goldberg, Ira
Hoover, Paul
Stern, S.

General Electric

Betz, E. S.
Cahall, R. B.
Christopher, G. F., Jr.
Clark, H. T.
Coombs, A. F.
Cooper, J. H.
DeMichele, D. J.
Dickerson, A.
Garland, N.
Germer, J. H.
Harmon, J. T.
Levine, Bernard
Macintyre, J. R.
Miller, D. R.
Oberholtzer, J. R.
O'Leary, J. J.
Porter, J.
Stephens, J. A.
Tharpe, R. L.
Weston, A. M

General Mills

Hedin, R. S.
Neumeier, K. E.
Sandgren, M. A.

General Motors

Anderson, J. O.
Aronson, C. A.
Burns, J. R.
Danse, L. A.
Gallant, R. A.

General Tire & Rubber

Sill, H. D.

Gilfillan Bros.

Hollowich, G. G.

Glass Fibers

Hadley, J. B.
Musikant, Solomon
Temple, W. M.

Goodyear Aircraft

Funk, R. W.
Rice, J. E.
Weiss, S. I.

Grumman Aircraft

Houghton, R. A.
Sheeves, A.

Gulton Mfg.

Dranetz, A. I.
Orlacchio, A. W.

Houdaille-Hershey

Henrich, R. E.
O'Connor, B. E.

Hughes Aircraft

Curtis, A. J.
Grosser, C. E.
Hutcheson, J. H.
Lobdell, F. W.
Morrow, C. T.
Steinman, J.

International Business Machines

Engstrom, J.
Koon, K.
Pattison, R. E.
Peters, A.

Jack & Heintz

Drlik, Martin
Hine, J. B.

Jeffersonville QM Depot

Rhodes, W. L.

Johns Hopkins University

Hoppmann, W. H.

Johns Hopkins ORO

Huffington, N. J., Jr.

Knolls APL

Wilbur, G. H.

Kollsman Instrument

Glaser, E. M.

Korfund

Harris, J.
Weindling, J. I.

Lear, Inc.

Altman, O.
Gorton, E. D.
Napier, Bernard
Odin, V. W.
Strodtman, Lee
Wortman, A. L.

Lord Mfg.

Anderson, J. R.
Billman, G. H.
Goodill, J. J.
Wallerstein, L., Jr.

Marine Corps

Mascho, H. F.

Marquardt Aircraft

Stoddard, C. P.

Glenn L. Martin

Borofka, F. J.
Eagle, E. L.
Kirchman, E. J.
Woodard, W. W.

Massachusetts Inst. of Tech.

Badessa, R. S.
Fincke, W. H.
Moore, M. H., Jr.
Zapf, K. L.

MB Mfg. Co.

Bligard, E.
Cottle, H. N.
Oravec, E. G.
Unholtz, K.

McDonnell Aircraft

Holliday, John
Ramey, M. L.

Wm. Miller Inst.

Dennis, P. A.
Downs, G. W.

Minn.-Honeywell

Clausen, W. H.
Soderholm, L.
Taylor, G.
Totten, G.

National Bureau of Standards

Ashley, R. B. (Corona)
Barclay, R.
Bayhi, F. H., Jr.
Blagman, Geo.
Camm, J. C.
Cooke, E. A.
Dobriansky, B. J.
Doctor, N. J.
Drake, John
Drane, W. C.
Edelman, S.
Fine, Aleck (Corona)
Garland, Wm.
Gelsleichter, W. C.
Hillstrom, W.
Hobbs, E. V.
Hull, Wilson
Jones, Earle
Kahn, Norman
Kaufmann, Jos.
Landers, H. M.
Lipnick, M.
Maki, K. K. (Corona)
McGinnis, R. P.
McNamee, B. I. (Corona)
Monroe, T. I.
Nelson, R. W.
Perls, T. A.
Ramberg, Walter
Reznek, Ben
Robinson, K.
Schoenberg, T. H.
Sensi, J. E.
Smith, E. R.
Smith, F. C.
Smith, R. W.
Smull, W. E.
Stansbury, C.
Weaver, P. R.
Willis, Ben
Witt, R. P.
Witte, Jos.
Yancey, J. D.

NACA - Langley

Youngblood, H. H.

NADC - Johnsville

Buterbaugh, F.
Chusid, W.
Clark, L. P.
Cohen, A. A.
Jones, R.
Nutter, R.
Shapiro, O.
Tait, J. N.

NAMC - Nav. Aircraft Fac.

Potter, Edwin

NAMC - Naval Air Exp. Sta.

Freed, Chas.
Greenland, M. G.
Schwartz, Samuel
Weiss, D. E.

Naval Air Station - NAMTC

Garcia, M. A.

NATC - Patuxent River

Ii, L. B.
Neil, R.
Wilson, R. T.

NEL - San Diego

Shipway, G. D.

NEES - Annapolis

Moeller, K. G. F.
Rekate, H. L.
Schloss, F.
Shovestul, P. J.
Smalzel, C. W.

Naval Gun Factory

Ballou, E. J.
Bryan, W. L.
Busch, M. F.
Carpenter, S. O.
Cohill, J. A.
Cook, L. L., Jr.
Gradijan, M.
Grubb, F. R.
Hamill, T. E.
Kossan, R. L.
Leventhal, A.
Long, O. M.
Lurie, W.
McKenzie, W. E.
Moore, W. L.
O'Keefe, J. M.
Sanders, W. R.
Scharff, L.
Schlachman, B.
Tocha, W. J.
Wilder, G. E.
Zimmerman, J. J.

Naval Medical Res. Inst.

Madelung, G. H.

Naval Ordnance Plant

Humble, John
Kuonen, C. E.

Naval Ordnance Plant (Cont'd)

Nabal, Emerson
Patrick, M. F.
Strobel, L. P.

Naval Ordnance Lab.

Armstrong, J. H.
Baldwin, A. W.
Bartos, J. S.
Bennett, R. D.
Brueggeman, W. C.
Butler, N. C.
Campbell, C. E.
Chathan, T. K.
Chaplick, R. G.
Crawford, M. W.
DeVost, V. F.
Diggins, J. L.
Fisher, H. L.
Fisher, J. C.
Gillean, M. E.
Luttrell, J. L.
Marlowe, D. E.
McClellan, H. F.
Mead, R. F.
New, J. C.
Obenschain, A. W.
Seaward, P. E.
Seely, R. E.
Smith, E. E., Jr.
Suhr, E. H.
Sullivan, D. E.
Sweger, W. J.
Talcott, J. W.
Tate, M. B.
Varhus, H. H.
Vogl, J. F.
Williams, A. H.
Young, J. D.

Naval Ord. Test Sta.

Alford, J. L.
Blatt, M. D.
Fairbanks, D. H.
Hart, L. W.
Hise, R. D.
Horine, C.
Judin, J. P.
Machowsky, John
Mapes, J. M.
Ott, Percy
Reynolds, R. W.
Taylor, H. A.

Naval Ord. Expl. Unit

Allison, J. M.
Deacon, N. E.
Miles, J. R.
Rafel, Norman

Naval Research Laboratory

Belsheim, R. O.
Beltz, W. H.
Blake, R. E.
Clements, E. W.
Drury, A.
Eney, H. E.
Fitzpatrick, W. N.
Forkois, H. M.
Graves, J. D.
Hardy, Virgil
Klein, Elias
Marcus, Henri
Malitson, H. H.
Minneman, F. L.
Parks, A. O.
Poole, H. S.
Shafer, E. J.
Rogers, G. L.
Russ, F. J.
Russ, R. G.
Sanders, W. H.
Trent, H. M.
Vigness, I.
Walsh, J. P.
Woodward, K. E.
Approx. 100 others for individual papers

Navy Supply R. & D. Facility

Akrep, J. P.
Kressen, W. C.

Naval Underwater Ord. Sta.

Murphy, F. G.

Navy Underwater Sound Lab.

Burnham, B. B.
Farrell, A. R.
Lattin, F. N.
Whannel, R. L.

New York Naval Shipyard

Knopfle, W. H.
Schnee, Marvin

North American

Ferguson, A. J.
Lothringer, R. C.
McDonald, B. A.

Northrop

Brown, W. H.
Cole, D. M., Jr.
Lenon, V. P.
Shwartz, M. N.

Office of Chief Signal Officer

Davison, G. E.
Smith, W. M.

Office of Naval Research

Crowley, J. M.
Liebowitz, H.

Army Ordnance

Hansen, A. B.
Lucht, V. F.

Owens-Corning

Armstrong, M. C.
Enright, W. T.
Heard, S. K.

Penna. State College (ORL)

Hausrath, A. H.
Marboe, R. F.

Philco

Bailey, E. P.
Binder, D.
O'Neill, J. V.
Sumerlin, W. T.

Picatinny Arsenal

Buck, G. R.
Gogliucci, A.

Portsmouth Naval Shipyard

Matheson, J. C.

Taylor, E. C.

Puget Sound Naval Shipyard

Watson, J. M.

QM Food & Container Inst.

Myers, E. C.

Queen City Tulatex Corp.

Underhill, A. M.

Radio Corp. of America

Carter, W. W.
Crawford, J. V.
Hayhurst, T. S.
Jacobs, L.
Lichtenberg, E. N.

Raritan Arsenal

Legare, M. G.

Raytheon

Dashefsky, E. L.
Kriby, W. B.
Weaver, F. W.

Reaction Motors

Borcherdt, W. O.

Redstone Arsenal

Bolon, H. C.
Hammett, C. E.
Hellebrand, E. H.
Hueter, Hans
Huston, M. E.
Jones, B. P.
Owen, E. L.
Pearson, J. C.
Shepherd, J. T.

Reed Research

Charner, S. A
Craig, D. R.
Price, J. F
Youngs, H. S.

Rensselaer Polytechnic Inst.

Macduff, J. N.

Research Corp.

Pierson, T. A.

Research & Development Board

Benthal, L.
Forsyth, P. S.
Gasdor, A. J., Jr.
Martin, M. R.
Selby, M. L.

Research & Development Board (Cont'd)

Thompson, L. T. E.
Ward, R.

Robinson Aviation

Lawrence, H. C.

Rome ADC

Carroll, J. F.
Catenaro, E. A.
Nares, H. E., Jr.

Ryan Aeronautical

Baxter, J. W.

Sandia Corp.

Adams, P. H.
Dupree, D. T.
Ellett, D. M.

Savanna Ord. Depot

Ehringer, A. G.

Signal Corps Eng. Labs.

Feder, Earl
Rice, Keith
Stout, H. L.

SC Plant Eng. Agency

Hiatt, E. D.,

SC Plant Eng. Agency (Cont'd)

Thompson, L., Jr.

Foster Snell, Inc.

Hardy, W. L.
McGill, J. P.

Sonotone

Johnson, H.
Klingener, H.
Myers, H.
Richman, S.

Southwest Research

Doak, R. A.

Sperry Gyroscope

Cope, A. P.
Hawkins, R.
Kelsey, A. S., Jr.
Muller, J.
Ward, R. B.
Yost, J.
Zaid, M.

Springfield Armory

Joachim, H. S.
Maier, K. W.

Statham Labs.

Motz, C. A.

Sunstrand Mach. Tool Co.

Straznickas, D.

Target Rock

Bergdahl, E. O.
Campani, J.
Crausman, M. A.

Taub Engineering

Heilprin, L.
Janeway, C.
Taub, Alex

W. D. Teague Associates

Ruddy, W. R.

Texas Instrument

Berkshire, R. H.
Congdon, R. L.
Darrouzet, J. L.

Univ. of Texas

Ripperger, E. A.

Thompson Products

Colling, J.
Gabacz, L.

Transp. R. & D. Sta.

Pleasants, D. S.
Vichness, J. A.

Vitro Corp. of America

Judson, R. B.
Kerley, J. J.
Violette, H. E.

Westinghouse

Friedrich, C.
Gibbons, J. F.

Weston Electric

Lamb, F. X.
Skidmore, W. H.

White Corp.

Fyfe, Clayton
Hafkemeyer, E. E.
White, H. L.

White Sands

Button, D. M.

Wright Air Development Center

Arnoult, J. E.
Birnbaum, S. M.
Bishop, A. B.
Cartnell, V. L.
Goldammer, F. R.
Golueke, C. A.
Grabner, C., Jr.
Kennard, D. C.
McDowell, C. W.
McIntosh, V. C.
Nielson, H. P.
Orensteen, R. B.
Ruther, F. J.
Scherberg, M. G.
Wernicke, B. K.

Welcome

Capt. W. H. Beltz, Director
Naval Research Laboratory

Good morning ladies and gentlemen. I am honored to welcome you to the Naval Research Laboratory and to the 20th Shock and Vibration Symposium. We are happy to have these meetings here occasionally because they remind us that the coordination of scientific knowledge is also a function of the modern research laboratory. This is especially true of these symposia which benefit so many agencies in the Department of Defense.

The first Shock and Vibration Symposium was held at NRL on 7 January 1947 under Navy sponsorship. Its purpose then was to collect, organize and disseminate all the research and development information in this field. Today, under enlarged RDB sponsorship, that is still its purpose.

I hope that your meeting here will be interesting and profitable.

Opening Remarks

L. T. E. Thompson, RDB

We are all acquainted with the importance of this series of symposia, conducted during the past few years, and the effects that have been felt by the collaborating organizations, which include almost all of the laboratories and centers in the country interested in weapon systems and systems in general in which shock and vibration effects are important.

We know that the Navy, and, in particular, the Naval Research Laboratory, has been largely responsible for much of the groundwork necessary for the success of this Activity.

I have several reasons for being pleased about this accomplishment. In the first place, I welcome the opportunity to be here, as a representative of the Research and Development Board, and to hear the large number of significant papers scheduled in the program.

In the second place, as one who was for a number of years part of a field center which benefited greatly from the activities of this group, I like what it is doing for field centers.

In the third place, I think we can see the beginning of a plan for the effective exchange of technical information between agencies working in the weapons program. Now, as in the past, we have to contend with what might be called a "communications problem." The solution which seems to be coming from your experience, and that of a few similar activities, may serve as a good pattern for facilitating the exchange of technical information in many of the Service research and development areas in this country. We know that some of the principal difficulties in effecting economy of effort and best technical progress are those of keeping up with what is being done by other groups working in a particular field, and of developing means for stimulating good contacts with the outside world.

As Captain Beltz has indicated, this organization really goes back to 1946, the time of the original directive by the Office of Naval Research, and to the first symposium, held in 1947. Since that time, nineteen of these meetings have been held. The activity has developed in a remarkable degree. The number of people who now attend these meetings averages three hundred. The published proceedings have a very wide distribution and great influence. I might mention here that approximately 900 people have obtained clearance to attend this present symposium.

In the last few days I have been looking over some of the documents which represent the proceedings of the earlier meetings. I noticed particularly the stimulating discussions in the reports that were given at the 18th Symposium. A great deal of the emphasis, directed toward the guided missiles field, was on reliability of performance. This is a subject which is now getting the increasing operations attention, developmentwise, it certainly deserves.

The 19th Symposium reviewed some interesting problems of the class, including the air-drop of materiel and the mechanics of parachuting.

I recall certain problems which used to worry us between the two world wars, and I am sure they are representative of problems that most of you have been concerned with at one time or another.

In those days we were impressed, for example, by the severity of the water-entry cycle for the nose of a bomb when it was dropped from what was thought to be rather high altitudes—we think of them now as moderate altitudes. These problems, having to do with shock loading, are still important in dealing with the design of penetrating missiles, including torpedoes and rockets as well as bombs. Much progress has been made in recent years in the study of this cycle.

The behavior of materials under ultra high-speed loading, the mechanics of penetration of solids at projectile speeds, and the behavior of metals under explosive loading, are old areas of continuing importance.

We have all been annoyed at one time or another with the behavior of optical systems in airplanes, and in other frameworks, which are vibrating or subject to transient acceleration. The problems associated with the precession of gyros, subject as they are to uncompensated moments as a result of this sort of loading through the bearing systems, and with navigation systems, which may be plagued with the effects of unsymmetric dissipative action in the coupling mechanism of the accelerometer or integrating gyro, so that even with closed acceleration cycles there may be spurious indications of displacement. Another problem area of interest is that of propellant grains under shock loading, since failure of the grains will result in serious effects.

Finally, I would like to come back to the point I mentioned earlier. I believe the Centralizing Activity has evolved and given exhaustive test to a procedure which seems to be one of the best pieces of machinery for developing perspective in technical areas involving many separated operating groups. The avoidance of unnecessary duplication of effort and the sharpening of the process of selecting important problem areas to be worked on are both reasonably expected consequences of the continuance of such a technical forum.

From the experience of this and similar activities, we may draw the conclusion that perhaps the best vehicle for the dissemination of technical information is an information agency associated closely with a laboratory or activity which is actually working in a particular field, combining its work with that of a sort of clearing house for technical information, such as the Armed Forces Technical Information Agency, which is jointly supported by the three Departments and operated by the Air Force. The necessary technical expertness can then be contributed in each of the fields covered without a large central technical staff.

Within the last few weeks Dr. Klein and I have had some discussion regarding the status of this Activity insofar as its relation with the Research and Development Board is concerned, and the desirability of reconfirming its position in that framework. Some of the discussion has been concerned with the possibility of identifying the Activity as a committee or a panel in the Research and Development Board structure. It appears now best not to pursue that subject until decisions have been made regarding the exact nature of the organization and the framework which will be the successor to the Research and Development Board. We are sure it is intended that there will be in the new organization a framework for insuring sound research and development perspective. Any facility required to enable the component activities in the weapons program to pick out the more important things to do, and the best techniques and associations for getting them done, will doubtless go ahead. I think I can say that the splendid work of this Activity for Shock and Vibration is likely to be vigorously supported because it does contribute importantly to the good focusing of an essential area of work.

* * *

DESIGNING FOR SHOCK - A STATUS REPORT

J. P. Walsh, NRL

The problem of finding a satisfactory system for designing equipments and structures to withstand shock motions is discussed. It is asserted that an easy, quick solution is not likely to be found. Existing methods are analyzed, and suggestions made for approaches to specific problems.

The designing of mechanical systems to withstand stresses due to dynamic loads has received a great deal of attention from both researchers and practical designers. Research workers are attempting to devise methods that the practical designer can use. It is a fair estimate of the situation to say that, so far, the practical designer is pretty much dissatisfied with the results of the research.

It is not my purpose to defend research results nor to apologize for lack of results. Rather I intend to describe the situation as I see it so that designers, particularly those in charge of design work, will understand what methods now are available to them and what they have to do for themselves to improve their situation.

The elastic stresses due to dynamic loads on a multi-degree-of-freedom system can be calculated. Almost all practical systems are multi-degree-of-freedom systems, and are characterized by having from two to an infinite number of degrees of freedom and a corresponding number of natural frequencies of vibration. Each natural frequency has associated with it a distribution of vibration amplitudes called the normal mode shape. When a structure is excited by a dynamic load, it vibrates at all its natural frequencies. The degree to which each natural mode is excited depends upon the excitation.

In order to calculate the elastic stresses in a multi-degree-of-freedom system, one must know the frequencies and the shapes of the normal modes of vibration and the stress distribution associated with each normal mode. Then the degree to which each normal mode is excited by the particular excitation under consideration must be computed, by finding the shock spectrum of the excitation. Last, in order to get the total stress, we must sum up the stresses associated with the individual modes. This method is used, for example, to compute the stresses in aircraft wings due to dynamic loads. The stresses in a uniform cantilever beam, due to a velocity change of its foundation, have been computed by this method, and excellent agreement of the exact solution with experimental results has been found. So a method does exist for computing the shock-induced stresses in a multi-degree-of-freedom system. But if this is true, what is the so called "design problem" all about?

The design problem involves finding another solution to the problem. We simply do not like the procedure outlined above, because it is an extremely complicated and expensive method of computing stresses in practical systems. I asked representatives of the Bureau of Aeronautics how much it costs, starting from scratch but knowing the excitations, to design a wing, including a computation of stresses due to landing and gusts. The best available estimate for this design job was, roughly, between a quarter and a half million dollars.

Although an exact solution to our problem exists in principal, it is extremely difficult and time consuming to carry out the necessary calculations. The problem that we are faced with, therefore, is to find approximations which are fast enough to be economically feasible, accurate enough to be of value to the practical designer, and simple enough so that the procedure can be understood and used by anyone with a background in statics.

There are at present two approaches used in designing for dynamic loads. The first of these

is the simplification method. In this procedure, a complicated system is reduced to a simpler system with fewer modes of vibration. The less complex system is then analyzed and exact answers are calculated for it. The simplest example of this approach—and certainly the most spectacular—is when a complicated system can be approximated well enough for this purpose by a single-degree-of-freedom system.

This method does not fulfill all the requirements of the ideal. Unfortunately, the person using it is required to have a background in dynamics. Recently, we have had some first-hand experience with the problem of simplification in designing equipment mock-ups to be used in a model submarine. These mock-ups represented actual full-scale equipments, and it was required that measurements made on them predict corresponding values on the prototype. This requirement, of course, is the same as requiring the calculations based on the simplified system predict results on the real system, and, from a technical point of view, is perhaps the most difficult part of the model submarine program. Professor Newmark, in a recent paper entitled "The Analysis and Design of Structures Subjected to Dynamic Loading," discusses the problem of simplification. He says, "The simplification of the structure for the purpose of making a dynamic analysis involves a good deal of engineering judgment."

Another discouraging aspect of the simplification approach is that it seems impossible to write adequate instructions for simplification. It is not possible to tell a person who does not have a background in vibration theory or dynamics how to simplify a structure. One possible way to get around this problem, however, is to have design organizations include a dynamics group to study the complicated systems, simplify them, and outline the calculation procedures required. The results of these studies could then be given to the designers.

The second approach to designing equipment to withstand shock (the easier and the more common of the two) is the static, or the "constant-number-of-g" method. The procedure here is as follows. The designer is told that the equipment must be designed for the static loads that would result if all the material in the equipment had a density N times as great as it actually has. The factor N is usually expressed as some number of g's. Present specifications imply this design method, for they usually state that the equipment shall be designed for some given number of g's.

What are the advantages and shortcomings of this method? First, the advantages:

(1) It is easily explained and easily learned.

(2) It does not require a knowledge of dynamics.

(3) It makes the designer aware of shock.

(4) By picking a high enough level of acceleration in g's, equipment can be made rugged enough to withstand the excitations.

The disadvantages are:

(1) It produces inefficiently designed equipments, that is, many equipments are heavier and larger than they should be.

(2) The magnitude and distribution of stress computed by this method cannot be equal or geometrically similar to that resulting from dynamic loads.

The last point is illustrated by the results from one of the submarine shock experiments referred to earlier. The measured response of the equipment was a shock-excited vibration, with the first and second modes of vibration dominating. The acceleration measured was several times greater than the design condition of 15 g's. The actual stress distribution was completely different from the static stress distribution assumed when using the static method. Luckily, the measured peak value of stress was within safe limits, but it occurred at a point in the structure that could not have been suspected using the static method. The section that the static method predicted as critical was perfectly safe, and another predicted as safe was critical. Increasing the section found to be critical by the static method could be a waste of material and would actually increase the load on the really critical section.

This failure of the shock loads and stresses to be distributed like those for steady acceleration can be understood by considering the properties of normal modes. In a normal mode of vibration, the distribution of amplitudes can be such that some regions vibrate in opposite directions from other regions. These regions, of course, are separated by boundaries of zero amplitude called nodes. Now an equipment vibrating as a part of a ship structure in a normal mode may or may not contain a node. If it does not contain a node, the inertia load distribution may bear some resemblance to that assumed in a static-design acceleration analysis. For modes for which the equipment contains one or more nodes, however, the inertia loads on the equipment may be markedly different from those assumed in a static analysis.

There are cases where the static method is quite satisfactory, but these must be discovered by a study of the excitation and the equipment. In other words, a very special result of the simplification method is a case where a uniform acceleration is adequate.

In spite of its shortcomings, the static method is far better than ignoring the problem completely. This method produces shock-proof equipment, not because the stress calculations predict service stresses, but rather because of the effects obtained when a designer is asked to design for some number of g's. These effects are:

(1) The designer is not only aware that shock is a factor in the design; he is given a concrete number that he can use.

(2) He can make calculations of sizes of certain sections such as hold-down bolts, legs, etc. These sizes then influence the sizes of other parts of the equipment, for designers have an axiom, "It isn't right if it doesn't look right."

(3) Since he is now aware of shock, his background in statics will prevent undesirable arrangements of components such as large overhanging masses, large weights high up in an equipment, etc.

In summary, the problem of designing for shock is in this state: An exact solution exists for the elastic stresses in a multi-degree-of-freedom system acted upon by dynamic load. Unfortunately, we do not like the answer because, in practical systems, it is an expensive procedure to calculate these stresses. But if it is sufficiently important to know the stresses in a system, this method is available. For example, it sometimes is used for computing stresses in aircraft wings. The design problem, as it exists today, is really one of searching for approximate methods which are faster, cheaper, but sufficiently accurate for our purpose. A multi-degree-of-freedom system under shock presents a complicated problem, and it may be some time before a good approximate solution is found. By "good" is meant one that satisfies the requirements of speed and accuracy, and has the ability to transform the dynamics problem into a statics problem. Although the outlook is not promising, the work should go on. There are two design methods in common use today. The first, which is truly an approximate method, is the so called simplification method; and the second, more physiological than scientific, is the constant-number-of-g method.

The simplification and analysis method produces better answers. It is more sound and the designer can get as good an answer as he wants, depending upon the degree of complication (or lack of simplification) that he is willing to accept. It requires a fair degree of sophistication to make the decisions that must be made, and it is doubtful that instructions telling exactly what to do could be written for the general problem.

The constant-number-of-g method is fast, and defines the problem as one of statics, but it is highly inaccurate.

Which method should we use? The decision depends upon how much the right answer is worth, and upon the training of the persons who are doing the work.

DISCUSSION

R. H. Oliver, BuShips: Mr. Walsh indicated that the Bureau prefers the latter of his two methods for use in design. This is quite true. I wish to compliment Mr. Walsh on his well-prepared paper and the manner of presentation. However, I would like to ask Mr. Walsh a particular question. He stated that at the present time, in this method, we need to know the level of shock which is to be used, the number of g's for which the equipment should be designed. I would like to ask Mr. Walsh if, in his work in the past two or three years, he has come out with any better values to be used in designing than are presently included in the Bureau's specifications.

Walsh: Thank you, Mr. Oliver. The answer to your question is "no," because it is impossible, actually, to specify a number of g's which is correct for general application. There are special cases (e.g., with uniform acceleration) where the static method produces a very good approximation to the dynamic stresses; but in a general application, the best one can do is to raise the level of g sufficiently to establish the desired margin of safety. The equipment can be made sufficiently rugged to withstand almost any given excitation, but this method cannot be depended on to produce either economical equipment or equipment in which the stresses bear some

reality to the stresses under shock. It is almost impossible to get a better level of g which is applicable to varied types of structures.

Jesse Markowitz, Barry Corp.: I would like to know what these values of g are that are referred to in the specifications. I am not aware that the specifications give specific design values of g that are to be utilized. As I understand it, equipments must pass the shock and hammer test in order to be acceptable. Some suggested design values, I gather, are available. Can you please tell me what they are?

Walsh: These values, contained in shipbuilding specifications, are so-called levels of g. They are usually presented either in the form of a curve, or they are given in a tabular form.

I refer you to a book published by the Bureau of Ships about 1946 - the author is W. P. Welch - in which the curve is given. This curve is carried in many ship specifications. If you are interested in actual numbers, I am sure Mr. Oliver can fit you out with a set.

Markowitz: I believe your paper has been very fine, and I think it sets forth the picture very well. But from your answer to my question I understand that, actually, there are no standard specifications in which the number of g's is spelled out, such as 40 g's. There are curves and other methods of presentation that show where the g's will vary for different equipments but there is no definite level or number of g's specified for a certain type of equipment.

Walsh: May I refer you to Mr. Oliver?. He, I am sure, knows the real answer to your question.

Oliver: These curves which are put into specifications are primarily guides to designers for equipments which are too heavy to shock test. Now, we should have some basis for accepting equipments, shock-proof-wise, which are too heavy to be shock tested on machines. So these curves are put in for the purpose of getting the designer, first of all, to realize that he has the problem and, secondly, in many cases, to submit calculations involving the critical sections - or expected critical sections - proving to us that he has taken into account these various factors. Now you won't find these factors in 40T9 or MIL-S-901, which are special specifications. You will find them, probably, in the equipment specification itself or, as Mr. Walsh pointed out, in the shipbuilding specifications.

H. M. Forkois, NRL: I believe that for designing equipment some values of g are given, and I do recall some of them. Depending on the location of a particular equipment in a vessel, these values are in the order of 35 to 65 g's. But the levels of shock to which we test on shock machines are considerably higher. These machines have been calibrated. Their calibrations are available, and certainly will give you an idea of the level of shock to be expected for a certain type of mass loading.

We have experimental results indicating stresses produced in bolts when subjected to shock motions involving accelerations as high as 300 g's, so we do have these values stated in the literature.

Dr. M. G. Scherberg, WADC: Mr. Walsh, you have indicated that the g method is limited in possibilities. Is there any quantitative measure as to what the range of accuracy is? Just what can you expect of it?

Walsh: Whether or not the constant-number-of-g method is accurate depends first upon the excitation to which you are designing and, secondly, upon the characteristics of your equipment.

We ran into an example of this, recently, in a problem of designing a rocket to withstand handling damage. The mount frequency was such that the gross motion of the rocket on its mounts was far below any natural frequency in the rocket. Under these conditions, one can take the peak value and design for it. But if one has an equipment which has a natural frequency of vibration right down in the excitation range, then you can excite three or four modes. The case I mentioned in the paper, in which two modes dominated, is an example. In this case, the critical section of stress was moved from the place that was suspected, using a static analysis, to a completely different location. So there is no general answer to the question; it depends upon the analysis of the problem.

I will sum up my answer: If, as a result of studying the equipment, you can simplify it sufficiently, a constant number of g's may be used - but there is no pat answer, in my opinion.

Scherberg: Is there no summary of experience that one can put down which would give a guide on specific kinds of structures?

Walsh: I would say packaging is a good example, where one can use the constant-g approach. One certainly cannot use it, in my opinion, for equipment in submarines.

Scherberg: I see. Then there are certain classes?

Walsh: Yes, and these are discovered by analyzing the problem. I don't believe a blanket order could be put out.

CDR. K. S. Brown, BuShips: My comment is concerned with the importance of loading as pertaining to the type of response. It seems to me there is a realm, unexplored, where experience is lacking regarding responses without nodes. That is, the constant-number-of-g method may have more justification than is apparent when used to simulate the short-duration loads in a shock wave. If we include pulse loading, there are durations of loads such that responses without nodes may exist, and the constant-number-of-g method may have much justification.

Walsh: This brings us back to the symposium of January. Yes, Commander Brown, if one is designing for the excitation characteristics of a bubble pulse, your statement is completely correct. Our experience, however, has been that the highest stresses are due to shock wave loading, so we still are faced with the relatively high-frequency excitations which correspond to, or are comparable with, those associated with higher modes of vibration of equipments.

F. Mintz, Armour Research Foundation: In your presentation, Mr. Walsh, you indicated that there are basically two extremes in the approach to the design of equipment for shock. Now, I have a method of design for shock which, I will admit, is based largely upon exposure over a period of years to many of your ideas on the subject; and I think it represents some sort of intermediate ground between the two extremes you discussed here. In this method, the shock spectrum is analyzed to determine the corresponding values of g for those particular frequencies to be considered in the design. Now, it seems to me that this method is considerably simpler than the one you employ for the exact method, and that it is considerably more accurate and also more complex than the constant-g method.

I would like to know what sort of excitation you design for - for example, in submarines or any given vehicle - when there is a rather wide variety of possible excitations?

Walsh: Mr. Mintz, the method which you have outlined is essentially the simplification method. As I said, one takes the complex situation, finds the modes, selects one or more of the first few modes, and then computes an exact solution for modes chosen, using the shock spectrum as defining the excitation - and this is precisely what I have chosen to call the "simplification method." One takes a complex situation and tries to get to the root of the situation; but one still ends up with, in a sense, an exact calculation for the simplified case.

Your question was, "What excitation does one use?" This is a very difficult question to answer because the amount of field data that we have on ships and vehicles is limited, although it is continuing to accumulate. People are able to draw up spectra for submarines, surface ships, and this sort of thing, and to provide bands of spectra to which designs are to be made. The question of whether or not one designs for the maximum or the minimum of a band is an operational-analysis-type question. The answer is determined by how much of a gamble one is willing to take.

F. C. Smith, Bureau of Standards: I'd like to make one comment on Mr. Mintz's statement, and I'd like to ask Mr. Walsh a question.

We are in a position at the Bureau—at least I am—of being a sort of go-between for dynamic theory and the design engineer. I have a formula, a recipe for designing against shock, which I would like to have Mr. Walsh or anyone else comment on.

We use the simplification method, except we do it this way. We find, by measurement or calculation, the lowest natural frequency of the equipment and forget the rest of it. We say, "Here is a shock picture, which, in our case, is usually igniter shock from the rocket fighter and this shock may have a duration of, let's say, ten ms. Now, if the period of the lowest natural mode and the period of the shock are such and such, then we are all right."

For example, if the pulse is very long compared to the period of the lowest natural mode of vibration—say three to one, or longer—then we use the static method for calculating stresses. This reduces the shock spectrum to one simple pulse and reduces test mechanisms to simple one-degree-of-freedom systems.

I feel that this is a pretty good system if you want one which is a little more refined than the static. It makes the design engineer conscious of shock. He realizes that if he has a shock of so many ms duration, and a structure with the period of the lowest natural mode of vibration in this region, he is going to have trouble.

The determination of natural frequency is usually made experimentally, and I think this is practical

because you almost inevitably have prototypes, at least on smaller things. Not for submarines, perhaps, but for less expensive items you have one or two prototypes that may be used to determine the lowest natural modes. The frequency of the lowest natural mode that is predominantly excited by the applied force is considered the most important critical frequency of the structure; hence the other modes are neglected. In the analysis of field data, if the ratio of duration of shock to period of structure is three to one or greater, conditions are considered to be satisfactory; whereas, if this ratio is about equal to one, then we start worrying. Now this may be a little crude, but it is more refined than a straight g value and it makes the designer conscious of shock. It think it would be more applicable for smaller components, such as missile fuzes or the missiles themselves, than for submarines. I would like Mr. Walsh to tell me what he thinks of this method.

Walsh: Well, I am delighted to see that you are using the simplification approach. I do not wish to imply that one has, in all cases, to worry about higher modes of vibration. I am simply stating examples where we did.

What you have done, essentially, is to analyze the problem and then make your decisions. You have simplified the problem to some extent and then you have passed it along. Now, another person working on the same problem might not make the same decision. He might say, "I am going to worry about the second mode." But it is the simplification method, nevertheless.

Charles E. Crede, Barry Corp.: Most of the comments have been directed toward the method discussed later in the paper, overlooking some of the implications in the first-mentioned analytical method for working on shock problems. I think most of us will agree that, if we know enough about the structure and the excitation, we can calculate the stresses and determine the maximum stresses in the structure. Thus it is implied that we know what stresses to design to. A number of additional questions are introduced because if we use the static properties of the material to calculate a presumed maximum design stress, we usually find out that the structure apparently can withstand higher stress dynamically than it can statically. There are a number of possible explanations for this structural property that, to my knowledge, have never been fully proved.

A structure excited by transients will vibrate for quite a number of cycles until the vibration dies out, so that the stress is repeated at various amplitudes until it reduces to zero, and a fatigue consideration is introduced into the problem. Now the first of these considerations, that of the apparently higher dynamic strength, tends to give a conservative design, whereas the latter tends to give an unconservative design; and the two may not cancel out in such a way that the analytical method gives the desired stress.

I wonder if Mr. Walsh has a comment on this observation?.

Walsh: Mr. Crede has raised two tough points. First, he has asked about raising the apparent yield point of materials. The second question concerns the fatigue problem.

I am not a worker in this field. The information that I have, however, is that yield-point raising can be factored into the design, using the best numbers that the workers in the field have produced. If desirable, of course, this change can be carried as a safety factor of the order of 10 or 20 percent.

On the question of fatigue, I have the opinion that if a material is below the yield stress, the relatively few cycles experienced under shock will not cause fatigue failure of the material. Does someone who works in this field know the answer to this problem?

Dr. H. H. Bleich, Columbia University: Concentration on fatigue consideration does not show any results different from the static loads over 100,000 load cycles and above, so that the order of shock cases does not get into the relative count at all.

* * *

THE COMPATIBILITY OF SHOCK SPECTRA WITH SHOCK DESIGN FACTORS

R. D. Ruggles, DTMB

The relationship between shock spectra and shock design factors, or design accelerations, which are currently in use in shock specifications for shipboard equipment, is discussed. It is shown that shock spectra are compatible with shock design factors and can provide a useful complement to design specifications.

There are, in general, two simplified approaches to the problem of specifying shock design levels for shipboard equipment. One is based on shock design factors, or design accelerations, which vary according to the weight of the equipment; the other is based on shock spectra, in which the design accelerations are a function of the natural frequency of a component of the equipment. The two methods will be discussed in an effort to show under what circumstances they are compatible and to suggest a simple way of including both methods in design specifications.

With the method of shock design factors currently in use in Navy specifications, a single design factor is specified for a given weight of equipment. This factor is multiplied by the supported weight to obtain a static load for use in designing the strength members. If the equipment is sufficiently rigid, all parts of the equipment will obtain essentially the same maximum bodily acceleration when the base of the equipment is subjected to a shock motion; and when the weight of the equipment is sufficiently large relative to the weight of the supporting ship structure, it will modify the shock motion of the structure and tend to reduce these maximum bodily accelerations. Consequently, for rigid equipment, specifying design factors as a function of the equipment weight is a reasonably valid approach.

This immediately raises two questions: How rigid must the equipment be before a single design factor is valid? How does one specify design levels for less rigid equipment? An approximate answer to both these questions can be provided by shock spectra. By analysis on the basis of one-degree-of-freedom systems, the method of shock spectra simplifies the problem of assessing the effect of resilience or flexibility. Although all physical structures possess many degrees of freedom, one of them is frequently so predominant that, for practical purposes, it determines the behavior of the system. Consequently, many simple structures may be idealized by one-degree-of-freedom systems with sufficient accuracy to estimate the required strength of the structure to resist shock.

For design purposes, the maximum response of the one-degree-of-freedom systems is of interest. For a given shock excitation, if the maximum response of a number of simple systems, each having a different natural frequency, is determined and this peak response is plotted against natural frequency, a shock spectrum is obtained.

The relationship between shock spectra and shock design factors as a function only of equipment weight can best be discussed by first considering the actual shipboard shock motion. The magnitude and characteristics of the shock motion are influenced to a considerable extent by the type of ship, the location within the ship, and the size and position of the charge, as well as by the weight of the equipment at the test position. During underwater explosion tests, measurements taken of the motion of ship structures supporting equipment or simulated equipment generally show an initial rapid rise in velocity to a peak value, followed by a damped oscillating motion. The acceleration associated with the initial rise in velocity is generally the maximum value which occurs in the shock motion.

If the response of simple mechanical systems to the shock motion is determined, the resulting shock spectra will indicate the maximum accelerations which would be obtained by simple systems of various natural frequencies. These values can be plotted as a ratio of the acceleration of the flexible systems to the maximum acceleration associated with the exciting motion. Spectra plotted in this form can then be interpreted more readily. Generally, there will be a low-frequency range where the acceleration ratio will be less than unity, i.e., where attenuation of the shock excitation acceleration would be expected and resilience is beneficial; there will be a range of frequencies where resonance effects will increase the acceleration and the ratio will be greater than one; and there will be an upper-frequency range where for relatively rigid systems the response will be essentially the same as the exciting motion and will have the same maximum acceleration. These acceleration-ratio spectra are analogous to the transmissibility curves found in vibration literature, differing primarily in the fact that the exciting function is a transient rather than a steady-state motion.

If the shock design factors given as a function of equipment weight are based on the estimated maximum shock accelerations to which the equipment will be subjected, they will provide design levels for rigid equipment. A representative shock spectrum plotted as an acceleration ratio will provide the additional information necessary to determine: (1) the limiting frequency above which systems can be considered rigid and consequently designed to the specified shock design factors; (2) the range of frequencies where amplification may occur; and (3) the reduction in the shock design factor which may be permitted when designing low-frequency components.

The feasibility of specifying shock design levels by the method suggested above presupposes several conditions. In the first place, a knowledge of the maximum shock motions to which shipboard equipment may be subjected is necessary. Secondly, the shock design factors for rigid equipment should be based on the maximum accelerations associated with the shock motion. Finally, it must be possible to select suitable shock design spectra.

Information on the maximum shock motions to be expected is, of course, very important. Although the available data are far from complete, reliable estimates can be made in some cases, and part of the Navy's shock program is directed at obtaining additional information. In any event, specifying realistic shock design levels, regardless of the method used, requires prior information on the shock motions to be expected.

With an estimate of the maximum shock accelerations, shock design factors for rigid equipment may be determined. For the most conservative design, these design factors could be made numerically equal to the maximum accelerations in g's. This is generally impractical, however, since the maximum accelerations are of such large magnitude. Therefore, various mitigating factors must be considered in order to reduce the design factors. Among these are the effect of delayed yield and the acceptability of a small amount of permanent set. For the purpose of the present discussion, the important point is that the reduction should not be completely arbitrary. If it is clearly understood why the accelerations have been reduced to obtain shock design factors for rigid equipment, then it should be possible to modify the shock spectra accordingly in order to obtain appropriate design spectra.

The problem of selecting representative shock spectra is not a simple one, but it is not insurmountable. A study of the shock motions and the resulting shock spectra encountered at various positions on a particular type of ship generally reveals a more or less systematic pattern. If shock design factors are given as a function of the general location in the ship, as well as of the weight of the equipment, corresponding shock design spectra can generally be selected which will give a fairly reliable indication of the effect of resilience.

In order to appreciate more fully the significance that shock spectra might have in design specifications, typical shock motions and shock spectra obtained on two types of ships are reproduced in Figures 1 and 2. It should be noted that

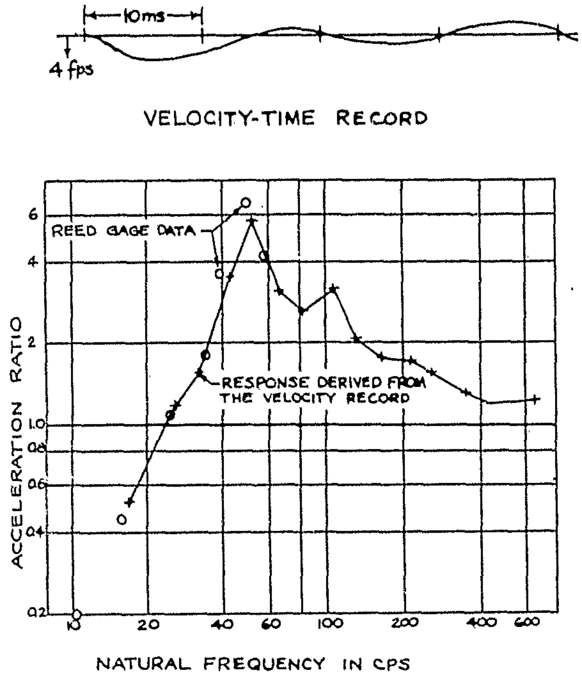


Figure 1 - Shock data obtained on a destroyer at a test position loaded by a 2240-lb weight

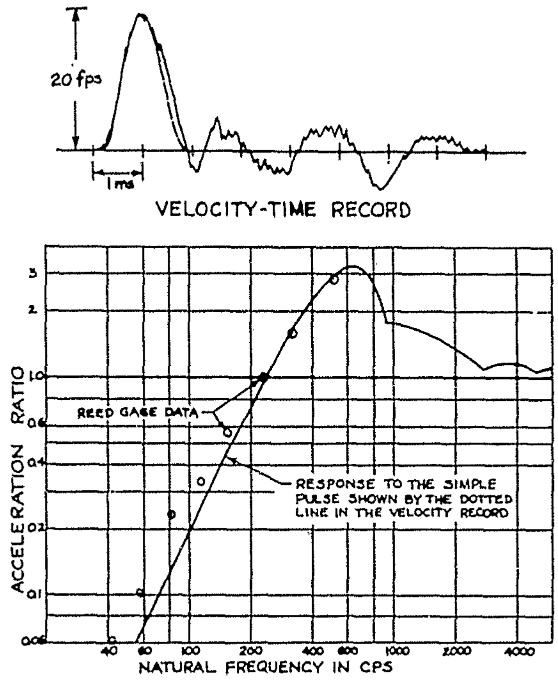


Figure 2 - Shock data obtained on a submarine hull at a test position loaded by an 8000-lb weight

these shock motions do not represent the maximum magnitudes. Figure 1 shows the results for a position on the bottom of a destroyer loaded by a one-ton weight. For such a position it is apparent from the spectrum that systems whose frequencies are above about 400 cps probably can be regarded as rigid systems, while systems of less than about 25 cps will receive less than the maximum acceleration of the shock motion. In the range of frequencies between these two extremes, there will be amplification which is appreciable in this particular case, because of the comparatively undamped oscillations of the shock motion.

The results shown in Figure 2 were obtained for a position on the side of a submarine hull loaded by a four-ton weight. In this figure, the plotted points in the spectrum represent reed-gage data, and the smooth curve is the response to the simple analytical pulse of velocity shown by the broken line on the velocity-time graph. Since the duration of this velocity pulse is very short, the resulting spectrum is significant over a greater range of frequencies than is the case for the destroyer curve. For such a position on a submarine, it is apparent that systems must have rather high natural frequencies before they can be considered rigid. Conversely, however, any frequencies less than about 200 cps will attenuate the shock motion. From this example, it is obvious that resilience will be an especially important factor in the design of submarine equipment.

It should be emphasized that the methods discussed in this paper are quite elementary and are not new. The basic idea is similar to that used in the Bureau of Ships Shock Design Manual. Furthermore, it is not offered as a cure for all the problems of designing to resist shock. The difficulty of applying the design data in actual design problems still remains. The important point is that shock spectra are significant and should be incorporated, in some form, into design guides or design specifications. The idea of using constant design factors for rigid components and acceleration-ratio design spectra for resilient components is suggested as a first step in this direction.

DISCUSSION

Fred Mintz, Armour Research: There is one question I would like to ask about the second curve. You plotted acceleration ratio against frequency. What is the reference acceleration? You end up with an acceleration ratio of about 3, with a velocity of 20 ft per sec. The accelerations must be considerably higher than those used for the other curve.

Ruggles: The base for the ratio is the maximum acceleration associated with the shock motion shown by the velocity time record. In this particular case, the maximum was 600 or 700 g's or larger, so three times this value is rather high. I avoided actual numbers because I did not want to leave the impression that this number of g's is representative of shock on submarines.

Mintz: There is one point about the shock spectrum in the second curve that, I think, should not go unnoticed here. The curve represents essentially a constant velocity shock, up to a frequency of about 400 cps. In this kind of shock, acceleration varies directly as the natural frequency. This is a clear case where a good shock isolator with a relatively low natural frequency would sharply decrease the accelerations sustained by the equipment.

I wonder if Mr. Ruggles has any comment on the effect of isolation over the frequency range that I have referred to.

Ruggles: I would like to comment on it for two reasons. Firstly, I fully agree that shock isolators will have definite advantages in that region.

Secondly, I would like to point out that the variation of acceleration is not exactly like that obtained by a simple step velocity change. The variation is more nearly proportional to the square of the natural frequency than to the frequency itself. This is primarily due to the fact that the excitation is a pulse involving two changes in velocity, from zero to the peak, and back down to zero again. However, it is quite true that for the type of shock loading considered, shock isolators should be very valuable.

DISCUSSION OF TRIP TO U.K. TO WITNESS SHOCK TESTS AND INSPECT TESTING FACILITIES, by H. L. Rich, DTMB

During the three-week period from 16 March through 2 April, representatives of the Bureau of Ships, the Engineering Experiment Station, the Underwater Explosion Research Division, and the David Taylor Model Basin were taken on a whirlwind tour of ten Admiralty Establishments concerned with shock in ships. I will briefly summarize the items discussed and viewed at several of the establishments.

A primary purpose of the trip was to witness the two final underwater explosion shock tests conducted by the Naval Construction Research Establishment in Scotland against the destroyer HMS Oudenarde. Consequently two visits were made to this establishment located at Rosyth, Scotland. During these visits we inspected the target vessel before and after each shot, examined the recording instruments on the target and on the recording vessels HMS ENDSLEIGH and the TRV-6, and observed the procedures. A summary of the results of all shots on this vessel was obtained.

In addition to the tests on the OUDENARDE, we were conducted on a tour of the establishment during which we saw work on current NCRE projects. The tour was followed by a series of discussions during which questions were answered and results of recent work were reviewed.

Facilities Seen at NCRE

(1) The new 500-ton capacity static test frame for testing large structures. At the time of our visit a 1/2-scale section of an aircraft carrier deck was being assembled for test.

(2) The metallurgy lab in which research is being carried on in brittle fracture and fatigue of metals.

(3) The submarine test tank in which submarine models are hydrostatically tested.

(4) The setup for testing superstructure and funnel models designed to withstand an A-bomb blast.

(5) The setup for testing automatic flapper valves for ship funnels. These are designed to prevent quenching of boilers by an A-bomb blast.

(6) Facilities set up to determine the effect of blast from jet launchings on decks, bulkheads, and topside equipment of ships.

(7) The laboratory where equipment to determine the bending strength of metals under shock load is set up.

Subjects Discussed

(1) Results of previous underwater explosion tests on naval vessels. In particular two reports being prepared by M. J. C. Lawrence summarizing the results of all British full scale tests on submarines and destroyers. From the point of view of shock in submarines, the most important result stated is that the lethal radius of a charge does not increase rapidly with depth of submergence of the vessel. Consequently, if the radius for major equipment damage is much greater close to the surface than the radius for hull destruction, it would likewise be greater at operating depth.

(2) Shock instruments.

(3) High speed motion pictures showing effects of underwater explosions on nylon rubber being considered for use in sweeps for pressure-sensitive mines.

(4) Model testing vs. full scale tests.

(5) Advantages of internal vs. external framing for submarine pressure hulls.

(6) The new shock machine being designed in England (15-ton capacity). Plans for this machine are to be forwarded.

(7) Results of tests on HMS OUDENARDE.

A highlight of our trip was a visit to the Department of Naval Construction at Bath. This meeting was set up as a round table discussion of the shock problems being conducted by both Britain and the U. S. Seventeen representatives of eight British establishments attended this meeting. The particular detailed phases of the meeting included discussions of resilient isolation mountings, shock design of equipment and machinery, and effects of underwater explosions on ship structures.

General Questions Asked

The following were among the questions asked by the U. S. visitors and answered by the British representatives.

(1) What are your techniques and facilities for evaluation of isolation mountings?

(2) What are your vibration requirements and tests for machinery items? Are these expressed as allowable velocities, amplitudes, frequencies, etc. ?

(3) Do you use laboratory shock tests for evaluating mountings?

(4) What are the trends in policies related to noise isolation of machinery and equipment?

Judging from the answers to these questions, the British do not as yet have a specification which outlines the desired characteristics of mountings. They conduct a series of tests somewhat similar to those we make under our evaluation specification. Their mountings are not tested for as high static loads as ours, but the British feel confident that their mountings have sufficient mechanical strength to withstand shock. The noise problem is not of great interest except on submarines, but a great number of mountings are used for the purpose of providing shock protection to equipment on all types of vessels. In general, particularly for heavy items, very low frequency mountings (4 to 5 cps) are used. As a result the British are having stability troubles, especially in connection with diesel engines. They are attempting to develop relatively rigid stops to be used in conjunction with the mountings to improve the stability rather than to design mountings which provide such stops as an integral feature.

Questions Concerning British Shock Design Manual

(1) How does the design put the data presented into effective design?

(2) How far into the numerous components of the equipment are these data valid?

(3) Have any valid means been developed for reducing these data in magnitude for use in designing internal components of equipment?

(4) What account, if any, is taken on the flexibility of equipment items? In other words, how are inherent natural frequencies of items correlated with the design factors presented?

(5) How are shock requirements for equipment put into specification form?

(6) For equipments which are too heavy to be shock tested, what is required of the designer to prove he has arrived at a shock-proof design?

The answers to these questions indicated that the British are little, if any, better off in attaining a rational shock design method than we are. Designers are supposed to present calculations

which indicate to what levels of shock the holding down arrangements and main structural members are designed. These are checked and, if correct, the equipment is considered to meet the requirements for shock. Inasmuch as the British at present can shock test only items which weigh 600 lb or less, the calculation criterion is the basis of acceptance of the majority of equipment as regards shock resistance.

Electrical equipments which can be tested on the shock machine are tested rigidly attached to the machine. As a supposed additional factor of safety, the items are installed aboard ship on resilient mountings. As a result of this practice, sometimes failures result which did not occur when the item was tested on the machine. Also, trouble sometimes results from shipboard vibration. However, the British feel that in the long run the supposed added shock protection offsets the relatively few difficulties experienced.

In view of the fact that the British shock design factors are in general considerably higher than those specified in our late special and detail specifications, the following question was asked: Do you find that as a result of requiring equipment to be designed to these factors you experience any weight increase in your equipment? It was answered that in general the major change in design consisted in using better materials, and no weight penalty was experienced.

It is our belief that the exchange of ideas at this meeting was extremely beneficial to both the British and ourselves because the discussions cleared up points which have been in question for some time.

Visit to Chatham Dockyard

This visit was arranged so that we might inspect the hull construction, equipment, and equipment installations in British submarines. During the visit, two T-class conversions were inspected. The first was in drydock and was in the course of having a new hull section added. The second was almost completely outfitted. During our tour of the second vessel we observed British practices for resilient mounting of machinery and equipment. We saw examples of the use of "rigid-resilient" mountings, "W" mounts, and "ARL" mounts.

We noted differences between British practices as outlined by DNC and actual practices in the yards. As in this country, the need for better dissemination of information between design laboratories and shipyard was evident.

Visit to AML

The Admiralty Material Laboratory is a new agency set up for the centralization of studies of the properties of material used by the Admiralty. During our visit we toured the rubber laboratory which advises the various Admiralty organizations on the use of rubber and plastics including rubber for resilient mountings. The rubber lab has facilities for compounding the testing rubbers and plastics. Test facilities include equipment for static, dynamic, and environmental testing. In addition we visited one of the metallurgical laboratories and discussed impact tests of steel specimens.

Visit to ASRE

The function of the Admiralty Signal and Radar Establishment is primarily the development of electronic communication and radar equipment. The primary purpose of our visit was to discuss the mountings used for this equipment and to inspect typical installations of mounts. The mount commonly used by ASRE is the so-called "W" mount. This mount, manufactured for the Admiralty in several load capacities, consists essentially of two steel "U" brackets bonded to rubber. The arrangement is one in which the rubber is loaded in shear; consequently the stiffness of the mount is roughly constant in deflection. To demonstrate the quality of the rubber-metal bond in a typical "W" mount, a 100-lb mount was loaded statically to 20 times its rated load. At maximum load the elongation of the mount was approximately 1-1/2 in. with no evident deterioration of the load. The "W" type of mount is considerably different from the usual American mount in that it permits a considerably greater deflection under shock load for the same rated frequency. Consequently, it is frequently necessary to provide additional snubbers to stabilize machinery installed on "W" mounts.

Visit to AEL

At the Admiralty Engineering Laboratory we observed the conduct of static tests on resilient mountings under development. These included new strip type mountings based on the ARL design and "S" type mountings.

Visit to DEE

At the Department of Electrical Engineering the history of shock testing in Britain was reviewed and both the British 600-lb shock test machine and the submarine storage battery test machine were demonstrated.

CONCLUSIONS

I would like to point out here, some parallels and differences between design practices and design specifications here and in England. In view of the fact that there is now a movement starting to unify British and American shock specifications, this may be timely.

In effect, here and in Britain, we have the same type of equipment specifications covering any weight of equipment. In general, in both countries acceptance of a piece of equipment depends on whether it is light in weight and passes the shock machine test or is heavy and has been designed to a specified shock design factor. The primary difference between the practice in the two countries is due to the fact that we have a 4000-lb capacity shock machine, while the biggest British machine available at the moment is limited to the test of equipment weighing 600 lb or less. Consequently the acceptance of a much wider range of equipment in Britain depends on somebody being satisfied, without a test, that the item has been designed to the specified shock value. The British are now designing a new shock machine, which will have a capacity to test equipment weighing up to 15 tons. That capacity is about four times that of our largest machine.

As to the actual type of specification, in both countries the specification gives number of g's against an equipment weight. However, we have one specification which covers all ships and in general all types of equipment. The British have extended this a bit. They have two curves, one for Grade 1 equipment and the other for Grade 2 equipment. In addition to this, in order to take care of special cases, the British have a large number of additional curves which are classified Secret.

As far as the actual design is concerned, a method similar to that suggested by Mr. Ruggles is used by the British to take account of equipment resilience. Unfortunately details of the methods are contained in a Secret Manual.

A striking fact about British equipment is that although it is generally designed to shock design factors considerably higher than those in our specifications, the equipment weight is not increased significantly. British representatives say that with careful design and better materials acceptable equipment may be obtained with no increase in weight.

* * *

MEASUREMENTS OF ROCKET THRUST AT FREQUENCIES UP TO 4000 CPS

V. R. Boulton, Aerojet Corp.

Design features and some results from a stand constructed to measure high frequency components of thrust from a solid propellant rocket motor are presented. Thrust transient magnitudes and frequencies required by missile designers to establish criteria for component ruggedness, shock mounting, and rocket attachment fittings are discussed.

INTRODUCTION

This paper discusses the measurement of both transient and periodic variations in thrust in a solid propellant rocket motor. Although the considerations are general, the measuring system discussed herein is based on the sustainer rocket used with a particular missile. The information desired is an evaluation of the dynamic forces which originate from the rocket, since such forces may cause troublesome shock and vibration in the missile. This will provide another design parameter which has not previously been available.

To measure thrust, the motor generally is strapped to a static test stand which is restrained with a standard, commercially available type of load cell. With this method it is difficult to measure accurately variations in thrust which occur at rates higher than 100 cps. This limitation is imposed by the natural frequency of the system, composed of the stand, the motor, and the load cell.

Methods for successfully measuring chamber pressure in the rocket at frequencies up to 10,000 cps have been in use for several years. There have been no thorough attempts to correlate chamber pressure variations with thrust variations at these high frequencies by direct measurement. Excellent correlation has been obtained between thrust and chamber pressure in the frequency range below 100 cps, and it is believed that good correlation at higher frequencies should be obtained. However, this remains to be shown definitely.

BASIC SYSTEM

The initial objective of this program was to measure thrust variations at frequencies up to 10,000 cps. However, practical considerations led to a more conservative goal of not less than 2000 cps. The system, as designed, appears to give good results at frequencies as high as 4000 cps.

It is necessary to build a measuring system with natural frequencies somewhat higher than those to be measured, so that they will not interfere with the measurements. No particular difficulty is encountered in obtaining electronic amplifying and recording equipment with a flat frequency response from 0 to 10,000 cps. The mechanical part of the system, however, presents a greater problem, i.e., that of obtaining a system which is stiff enough to have the required high natural frequencies and at the same time has enough deflection to provide a measurable signal proportional to the thrust.

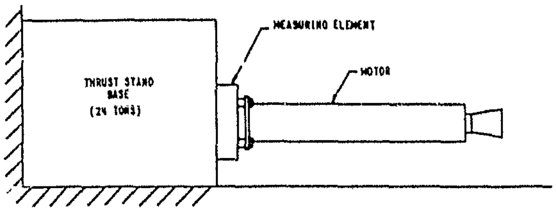


Figure 1 - Basic measuring system for high frequency thrust stand

The basic system is shown in Figure 1. The base of the stand, which is a large metal block mounted in concrete, provides a rigid surface against which the rocket thrust is exerted. The measuring element, or thrust transducer, is interposed between the block and the fore end of the motor. This measuring element must also be as stiff and simple as possible.

DESIGN

The thrust stand base is a 24-ton cast-iron "sow" block, selected because it is heavy enough so that the rocket thrust will not overcome the static friction. The block, which was purchased from a local steel-forging concern, is tapered so that the back surface transmits the force to the concrete wall behind it over a large area. The front surface of the base is machined to provide a smooth surface for mounting the measuring element.

The measuring element is a steel pad 16 in. in diameter, with a ring machined on it which mates with the fore end of the motor. Steel was chosen because of its high modulus of elasticity and good machinability. The large smooth surface between the base and the pad is necessary so that the stress, and therefore the resultant strain, at the interface be kept small. This is important if a high spring constant, or low strain per unit load, is to be maintained. The ring which is machined on this steel pad is the basic element of the thrust transducers; its strain is used as a measure of thrust. The ring has the same outside diameter as the motor and was selected in order to simulate the thrust attachment used in the missile.

Although the rest of the system was made as stiff as possible, the dimensions of the measuring ring, and therefore its stiffness, were determined by the sensitivity of the instrumentation to be used.

A thin cylinder about 3 in. long, threaded at one end and welded to the fore end of the motor, is normally used to attach the rocket to the missile body. On the motors to be used with this test stand, this cylinder is replaced with a very short and stiff ring with a highly finished surface which mates with the measuring ring. Lugs are provided on this ring for use in attaching the motor to the measuring element. The attachment is made by bolts and preload springs, whose function is to prestress the measuring element to the point where deflection is proportional to force.

The modification to the motor was made in full realization that the thrust variations measured would not necessarily be the same as those imparted to the body of the missile through the normal attachment cylinder. The forces produced by the rocket directly are the most useful to know; this configuration provides the system which measures these forces most accurately.

INSTRUMENTATION

Selection of the instrumentation system was based on the following considerations:

1. Frequency response (at least 0-4 kc)
2. Commercial availability of new instrumentation, or the use of types known to be satisfactory (it was not considered necessary or desirable to undertake an instrument development program)
3. Feasibility of use under the existing environmental conditions (high acoustic fields, corrosive gases, and remote recording)
4. Highest signal-to-noise ratio consistent with above.

Thrust is measured by evaluating the strain in the measuring ring by two methods:

1. A displacement transducer of the condenser type is mounted in the steel pad so that the deflection in the measuring ring produces a change in the air-gap spacing of the pickup condenser plates. A section of the finished surface at the fore end of the motor forms one plate of the condenser.

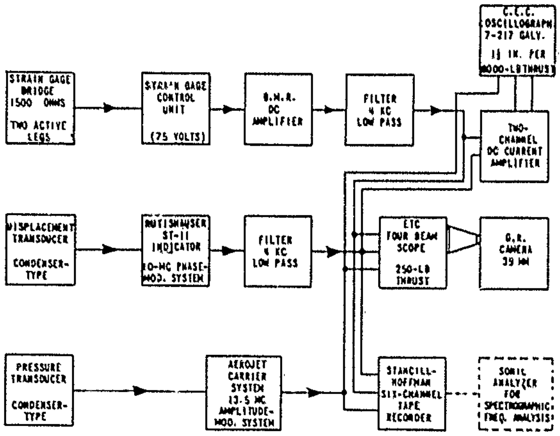


Figure 2 - Block diagram of instrumentation system for high frequency thrust stand

2. SR-4 strain gages of the bakelite type are bonded to the measuring ring. These are mounted to measure the average axial strain from four points on the surface of the ring. Temperature-compensating gages are also used.

With these systems the best noise level that could be expected was estimated to be equivalent to 1/4 microinch deflection for the condenser type transducer and 1/4 microinch/inch for the strain-gage system, using the best commercially available dc amplifier with sufficient bandwidth.

A block diagram of the instrumentation system is shown in Figure 2. The output of each of the three high frequency measuring channels (i.e., the strain-gage and displacement transducer systems and the condenser-type chamber pressure measuring system) is recorded by three different means:

1. A Consolidated Engineering recording oscillograph using 2000-cps galvanometers on which the static and dynamic data are recorded:

2. An ETC 4-beam oscilloscope and a General Radio 35-mm camera on which the dynamic components are amplified and recorded on an expanded time scale;

3. A Stancill-Hoffman 6-channel tape recorder. The magnetic tape record provides a means for analyzing the frequency components present by the use of a Panoramic sonic analyzer and General Radio sound analyzer. The 35-mm camera records provide good data on the amplitude of the variations in both thrust and chamber pressure over the entire frequency range. The oscillograph records show the static values, with variations up to 3000 cps superimposed.

Photographs of the instrumentation system used are shown in Figures 3 and 4.

DYNAMIC ANALYSIS

The criterion for determining the dimension of the measuring ring was to obtain sufficient deflection so that the noise level in the instrumentation system did not exceed the signal level for thrust variations of 1/2 percent of the static thrust level. This resulted in a ring 7/8 in. long with an area of 6 sq in.

With the design of the stand thus set, a dynamic analysis of its response was made in order to predict the usable frequency range. This analysis is based on the assumed spring-mass analog for the system (Figure 5). The differential equations of motion for the system, assuming no damping, are given below.

Figure 3 - Instrumentation system for high frequency thrust stand

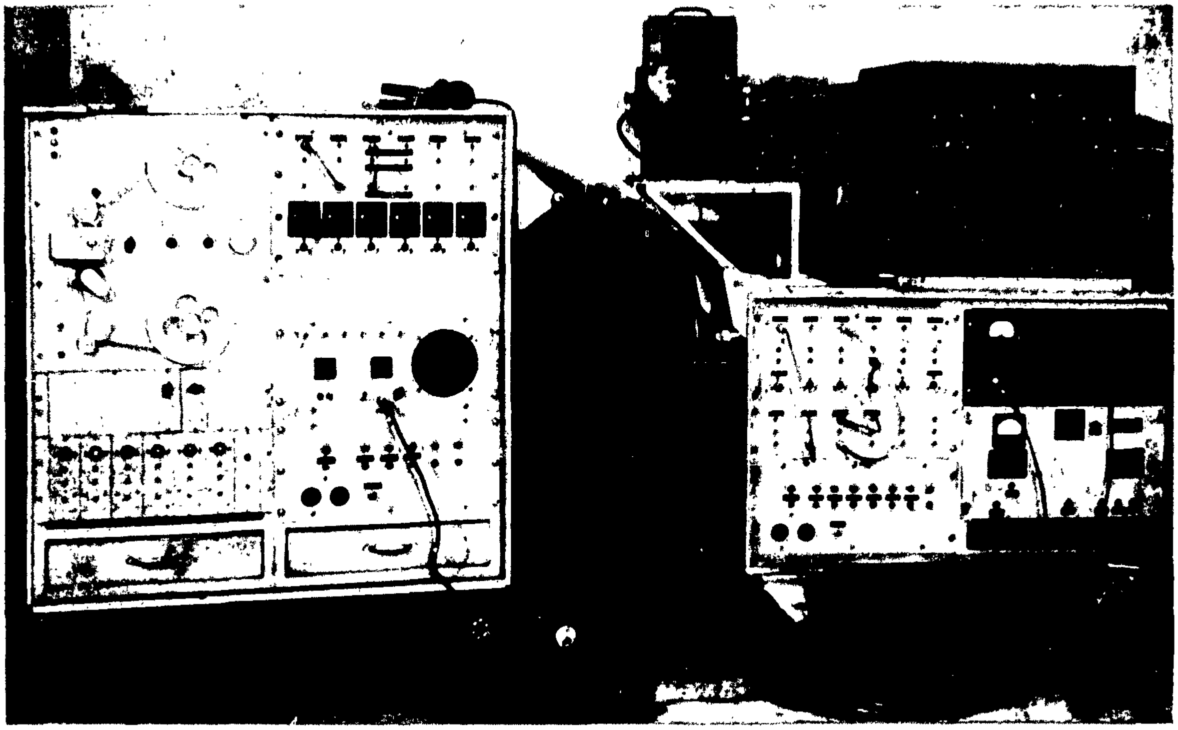


Figure 4 - Instrumentation system for high frequency thrust stand

$$M_1 p^2 X_1 = -K_1 X_1 + K_2 (X_2 - X_1)$$

$$M_2 p^2 X_2 = -K_2 (X_2 - X_1) + K_3 (X_3 - X_2) + F$$

$$M_3 p^2 X_3 = -K_3 (X_3 - X_2)$$

where

M_1 = mass of steel pad

M_2 = 1/2 motor mass

M_3 = 1/2 motor mass

K_1 = spring constant of steel pad and interface between pad and cast-iron base

K_2 = spring constant of measuring ring

K_3 = spring constant of chamber walls

F = forcing function

p = symbolic operator; $\frac{d}{dt}$ = iw for steady-state solution.

These equations are solved for the transfer function

$$\frac{X_2 - X_1}{F},$$

since this transfer function is the strain of the measuring ring per unit force. This is given in matrix notation below and the complete solution is shown.

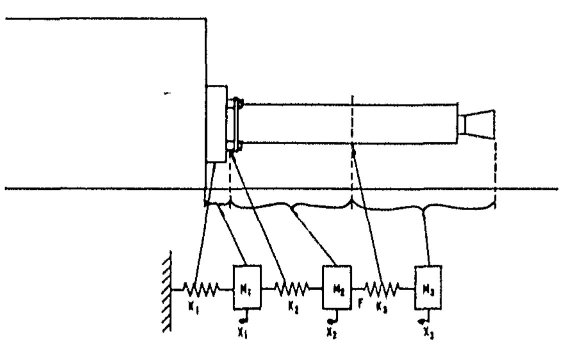


Figure 5 - Assumed spring-mass analog for the system

$$\begin{array}{cc} & \begin{array}{ccc} \underline{X_1} & \underline{X_2} & \underline{X_3} \end{array} \\ \begin{bmatrix} 0 \\ F \\ 0 \end{bmatrix} = & \begin{bmatrix} M_1p^2 + K_1 + K_2 & -K_2 & 0 \\ -K_2 & M_2p^2 + K_2 + K_3 & -K_3 \\ 0 & -K_3 & M_3p^2 + K_3 \end{bmatrix} \end{array}$$

$$\frac{X_2 - X_1}{F} = \frac{\Delta_{22} - \Delta_{21}}{\Delta}$$

$$= \frac{1}{M_2} \frac{W^4 - \left[\frac{K_1}{M_1} + \frac{K_3}{M_3}\right] W^2 + \frac{K_1K_3}{M_1M_3}}{-W^6 + \left[\frac{K_1+K_2}{M_1} + \frac{K_2+K_3}{M_2} + \frac{K_3}{M_3}\right] W^4 - \left[\frac{K_1K_2 + K_1K_3 + K_2K_3}{M_1M_2} + \frac{K_1K_3+K_2K_3}{M_1M_3} + \frac{K_2K_3}{M_2M_3}\right] W^2 + \frac{K_1K_2K_3}{M_1M_2M_3}}$$

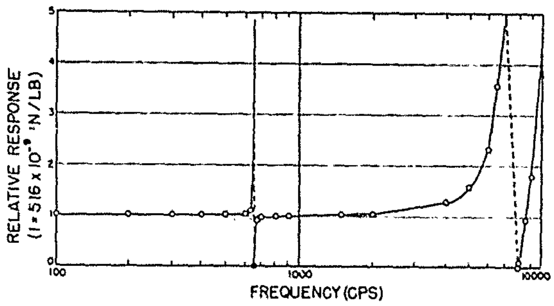


Figure 6 - Theoretical high frequency thrust stand response,

$\frac{X_2 - X_1}{F}$ versus frequency

The solution to this equation is shown in Figure 6 where the transfer function is plotted vs. frequency. The first resonance, occurring at about 650 cps, is a characteristic of the motor chamber and the motor mass; accordingly, it is necessary to measure any effects which it might have. The resonances at higher frequencies are functions of the thrust stand; consequently, the response of the instrumentation must be limited to a value somewhat below the second peak, which is at 7100 cps, so that the measured information may be representative of the thrust from the rocket.

It was anticipated that the actual resonant frequencies would be somewhat lower than the calculated values; since the damping was expected to be small, a conservative value of 4000 cps was taken as the estimated range of useful thrust data. Accordingly, low-pass filters with a cutoff frequency of 4000 cps were provided for use with both thrust-measuring systems.

DYNAMIC CALIBRATIONS

After the fabrication of the stand and the instrumentation system, dynamic calibrations were made in order to check the actual response of the stand. The first method used involved a steady-state type of dynamic calibration, in which an electromechanical vibrator was used to supply sinusoidally varying forces of approximately known values to the stand through the frequency range 0 to 10,000 cps. The results of this test on the complete system, including the motor, are shown in Figure 7. The resonance of the motor chamber, predicted to occur at about 650 cps in the analysis of the system, shows up in the dynamic calibration at 590 cps. The second harmonic is also readily discernible. In order to check the response of the stand without the motor a special calibration fixture was fabricated to provide a method of applying the calibration forces directly to the thrust-measuring ring. The setup used for this calibration is shown in Figure 8.

The second type of dynamic calibration used involved a "step" test, in which the preload bolts were tightened and a rod was attached to the fixture and pulled axially with a force less than the total preload. The rod was then sheared by

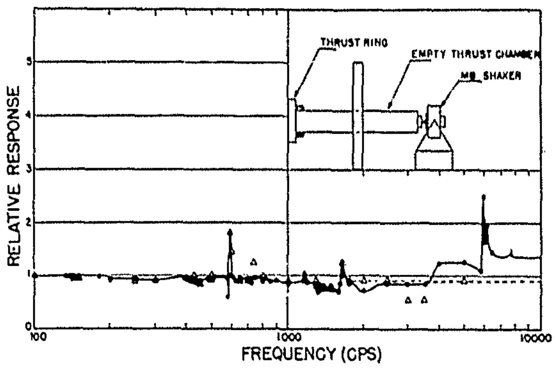


Figure 7 - Dynamic response of high frequency thrust stand

● high preload
△ low preload

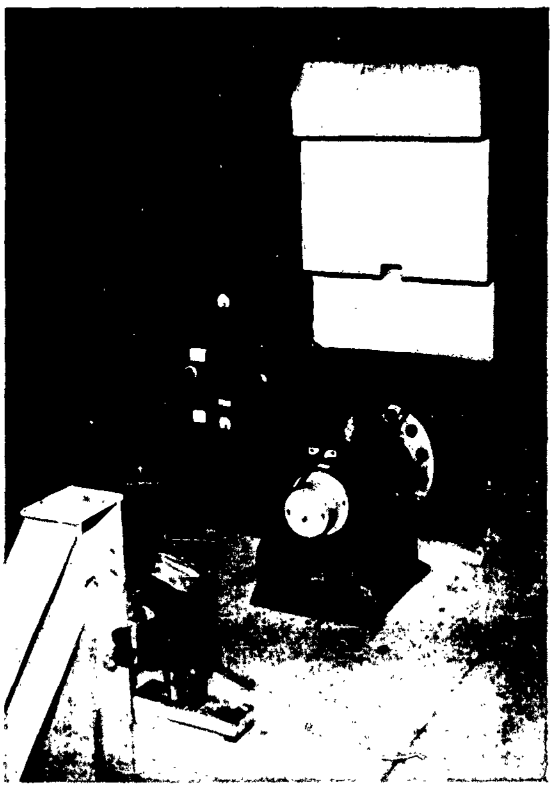


Figure 8 - Special calibration equipment

means of a blasting cap and primacord, resulting in the sudden application of an increment of force. The results of the "step" test using the calibration fixture are shown in Figure 9. The 3000-cps resonance which appears in the dynamic calibration, using the calibration fixture, does not appear when the motor is mounted on the

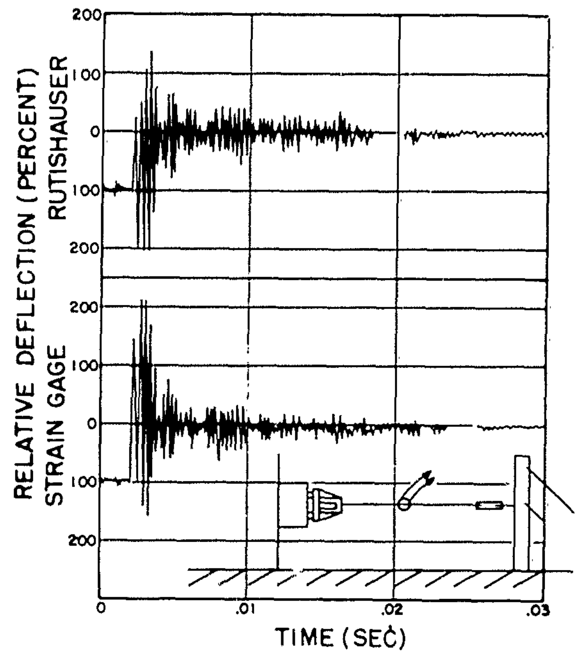


Figure 9 - Step response of high frequency thrust stand

stand. This resonance was therefore attributed to the calibration fixture, and is not considered to result from a limitation of the thrust-measuring system.

CONCLUSIONS

The results of these calibrations show that valid results up to at least 4000 cps can be obtained. It was found that extreme care was required in grounding the electrical components and shielding the equipment from acoustical fields in order to obtain the desired noise level in the instrumentation systems. Although this system was designed specifically for testing a particular rocket, it is readily adaptable to the testing of other solid-propellant rockets. It is necessary only to design and fabricate another sensing element to fit the attachments of the rocket to be tested.

Several test firings have been made using this test stand, and although the results are not yet available, it appears that satisfactory measurements are being made, thus establishing one more factor in the problem of missile shock and vibration.

DISCUSSION

A. S. Elder, Aberdeen: We are using strain gages to measure reactions on guns and have had difficulty in keeping a constant zero. Could you tell us how your instrumentation differs from the recommended techniques for recording similar data? This matter of technique is of considerable interest to us.

Boulton: The techniques which we used and which seemed to be quite successful are those recommended by a consultant in the field. There is a pamphlet on it; but I do not recall the title or author's name. However, I might say that we do have some zero problems due particularly to armature effect. Judging from the results so far, it appears that our study on thrust values will not be good primarily because of the zero drift. This does not involve the dynamic components in which we are primarily interested. We have other methods of measuring static components.

J. P. Walsh, NRL: Your diagram of the instrumentation implied that you take the signal from a pick-up, and then run it through some sort of spectrum analysis. I was wondering if you would tell us what the significance of the spectrum is under this type of excitation.

Boulton: We record the spectrum on tape and play it back through a sonic analyzer to get a spectrum analysis of the input during the time of firing. With the sonic analyzer, of course, we cannot get the time relationship between amplitude and frequency. However, we are able to get approximate amplitudes and fairly exact frequencies that occurred during firing by cutting the part of the tape out that represents the input just at the time of firing and playing this part through the spectrum analyzer many times.

Walsh: I am not familiar with the type of excitation that one sees under these conditions. Is it essentially steady state?

We have comparable problems in measuring vibration. Much of our data is recorded on paper, and then is run through a spectrum analyzer.

I wonder if anyone has concerned himself with this problem. What is the significance of a spectrum obtained when the excitation is not steady state?

Boulton: The excitation in some ways is essentially steady state. There are some transient types but, in general, due to combustion phenomena we do have sustained variations. I don't think I can give a better answer to your question. That is, for any type test the only thing that an analysis record would show would be the natural frequencies of any components affected by this type of excitation.

E. R. Smith, NBS: I would be glad to discuss the subject with Mr. Walsh after the meeting. We have spent about a year working on the problem, and have come up with some ideas, both theoretical and practical, as to what happens when you take a steady-state excitation and put it through a sonic analyzer.

J. Muller, Sperry: I was wondering if it is the same test as started originally. Are you trying to get an analysis of the spike type excitation of the missile motor during starting?

Boulton: Yes, it is the same test. We would like to get both spike type of excitation and any sustained type. I was merely saying that in any analysis, we would not be able to get a good picture of the spike type. However, we are able to see this sort of thing from the film record which does give us a time plot of the curves. This is one of the things we are interested in getting.

Muller: The reason this program was started was to see what effect the excitation has on the instrumentation or the instruments in the missile itself. Too bad we at Sperry didn't get to see the results.

Boulton: Probably the results are too late to be of very much help to you. However, I hope other results will be of more use.

Dr. E. H. Kennard, DTMB: A nonsteady disturbance can represent the relative amplitude of a harmonic vibration of a certain frequency, excited by that disturbance. That is, each point in the spectrum represents the response of a harmonic oscillator of that frequency starting from resonance at the time the disturbance starts. Won't that work, Mr. Walsh?

Walsh: Yes, that will work providing one has a tape which shows the whole story. The usual technique, as the speaker has pointed out, is to cut a portion of the tape out and subject this to harmonic analysis. Under these conditions I think my question still stands—what in the world does it mean?

Fred Mintz, Armour Research: We have had a little experience with the same problem.

Although I probably cannot answer Mr. Walsh's question, I would like to make some comments on it.

At Armour Research, we are interested in obtaining a time history of a transient pulse, in this case the output of an electronic tube we excited by shock, and we also wanted some information as to the frequency content. We tried to use the vibration analyzer put out by Kay Electric Company based on the Bell visible speech aid. I think the principle is sound because you can get your signal displayed in such a way that you know where the important peaks occur in the time history. Then you can select instants of that particular time of that history and get an instantaneous analysis at that moment.

I am not prepared to say how effectively the equipment operated. We were not entirely satisfied with some of the features. However, we did get some knowledge of the history of the frequency spectrum by that method.

I would also like to discuss briefly the resinous formula we have used for mounting Bakelite gages. I don't have the details with me although my group has used it. It is a rather complex formula. It depends not on baking, but on polymerization which takes place very quickly and seems to be quite effective in holding the gages under high strains and relatively high temperatures.

Dr. M. G. Scherberg, WADC: You stated that the applied forces in this case were steady state. In view of the question that has been raised about the instrumentation, how did you come to that decision? Is it a theoretical decision or from your instrumentation?

Boulton: If I said they were steady state I was speaking a little loosely, I think. The components we were analyzing were the periodic type of forces. The thrust of the rocket, of course, builds up from zero to full thrust in a short time interval, say a tenth of a second; goes along at roughly a steady value; and then falls off very quickly after burnout. During the build-up period there are some variations and also during the combustion period during the normal running, there are variations in thrust. As I mentioned, in the lower frequency range where we can see what is happening to the chamber pressure, we are able to correlate thrust and chamber pressure. Theoretically the thrust results from the chamber pressure; there are some things which might show up at one frequency and not at another, but by and large the chamber pressure gives a good indication of what the thrust is.

R. E. Blake, NRL: The problem of the meaning of the vibration analyzer output came up at NRL when we "analyzed" the vibration of a rocket. The cause of the vibration is evidently a series of random impulses due to variations of motor thrust and aerodynamic forces. The Fourier spectrum for random pulses, like that for any transient disturbance, is continuous and contains energy at all frequencies. This spectrum is modified by the rocket fuselage so that the vibration amplitude spectrum, while still continuous, tends to have peaks at natural frequencies of the fuselage. The proper scale to use in plotting a continuous spectrum is not clear.

On the other hand, a line spectrum of discrete component frequencies characterizes the steady-state Fourier spectrum. The amplitude scale of the Western Electric 3A Sound Analyzer used at NRL is established by inserting a known single-frequency sine wave and observing the amplitude of analyzer output. Thus the scale applies only to discrete line spectra in the analyzer output. These can be recognized as narrow spikes. Use of the scale for broader peaks which were on our rocket vibration records, is apparently an error.

It seems safe to say that the appearance of the spectrum analyzer output defines whether the vibration is steady state or transient. In general, one cannot say prior to this analysis whether rockets, freight cars, trucks, aircraft, etc. experience steady-state or transient disturbance or both.

* * *

CONTROLLING SHOCK AND VIBRATION BY FRICTION DAMPING

R. N. Janeway, Chrysler Corporation

General characteristics of viscous and constant friction or coulomb damping are compared to show where coulomb damping is advantageous. A new friction snubber is described, which approaches true coulomb damping by eliminating objectionable, high breakaway resistance usually encountered in friction devices. Practical applications to military tank and railroad car suspensions are given.

In the application of damping to vibrating systems, friction devices were for a long time the only available type. One of the earliest forms of friction damping was that inherent in leaf springs where the friction between the leaves was relied upon for damping. However, this is a rather special case because the friction surfaces also carry the load, and therefore, the frictional resistance tends to vary in proportion to displacement. Moreover, the degree of control is not flexible since only a lubricated condition can be relied on to maintain any reasonable constancy of friction or to keep surfaces satisfactory and operable over any extended period. Consequently, it became necessary to provide supplementary damping devices wherever adequate control was essential.

Until recently, dry friction damping devices have been generally unsatisfactory, both as to performance and life. These defects can be attributed to two principal factors: first, the use of metallic friction materials which have undesirable variations in friction coefficient as well as a high rate of wear; and second, a lack of precision in the relationship of the component parts which made close calibration of the friction impossible. in the meantime, the expanding need for suitable damping devices has been filled largely by various types of hydraulic mechanisms.

Aside from the practical deficiencies of previous friction devices, this type of damping has also suffered because its fundamental characteristics have not been thoroughly investigated. It is true that all textbooks on vibration refer in passing to coulomb damping, which is defined as damping by the action of a constant friction force in the system. On the other hand, viscous damping where the damping force varies in proportion to velocity, has been elaborately treated. Consequently, the fundamental characteristics of viscous damping are well established. Undoubtedly, the primary reason for this neglect of coulomb damping is the fact that it is exceedingly disagreeable to handle mathematically because it cannot be expressed as a continuous mathematical function. In contrast, viscous damping can be expressed as a continuous function in a linear differential equation and thus involves no frustrations to the mathematician.

Two comparatively recent developments have stimulated progress in the field of coulomb damping. First, a friction device is now available which embodies in practical form the ideal constant friction or coulomb damping. Consequently, a thorough knowledge of the characteristics of such damping has now become an objective of much more than academic interest. Second, the availability of electronic calculating machines has made it possible to solve quickly mathematical problems that defy longhand methods. This approach has already yielded valuable new information on the fundamental characteristics of both coulomb and viscous damping.

The subject matter of this paper is devoted largely to a comparative analysis of coulomb and viscous damping effects in vibrating systems of a single degree of freedom. Test results are also given for the practical applications of the new constant-friction snubber, which has already proved successful in several different types of service. However, the primary object is to present principles of damping which, it is hoped, will find useful application in a wide variety of problems involving the control of shock and vibration.

COMPARATIVE ANALYSIS OF VISCOUS AND COULOMB DAMPING

The following analysis is devoted primarily to vibrating systems of one degree of freedom, for two principal reasons:

(1) The characteristics of coulomb damping have not been fully explored as applied to more complex systems, and

(2) the practical applications to date, as well as many other possible applications, fall within this category.

However, the specific case of automobile suspension classified as a two-degree-of-freedom problem is examined to show why viscous damping is inherently better adapted than the coulomb type to the essential requirements of such a system.

In every case, the results are presented in terms of force transmission through the system. This is the significant measure of the protection afforded against shock and vibration by a suspension system. As the analysis will make clear, the amplitude of displacement with viscous damping is hopelessly misleading in any application where the transmitted forces are of primary importance.

Free Vibration

In all treatments of the subject one characteristic of coulomb damping is readily demonstrated, namely, that the reduction in amplitude of free vibration in successive cycles is a constant quantity,

$$\Delta X = 4F/K$$

where

ΔX = the reduction in amplitude per cycle,

F = the constant friction force, and

K = the spring rate.

Therefore, the damping effect is independent of the vibrating mass as well as of the magnitude of the displacement. In contrast, viscous damping of a free vibrating system is characterized by a constant amplitude ratio from cycle to cycle of the form

$$\frac{X_m}{X_{m+1}} = e^{\frac{2\pi b}{\sqrt{1-b^2}}}$$

where

b = damping factor, relative to the critical value,

X_m, X_{m+1} = displacement amplitude of two successive cycles.

The comparative effects of the two types of damping are best visualized by a force-displacement diagram, particularly because such a diagram graphically defines the integrated energy changes. As shown in Figure 1 for coulomb damping, starting from the initial amplitude of 1/2 in. the spring force and damping force are plotted separately against displacement. It will be noted that the ordinate scale is given in force units relative to K, the spring rate, and so the diagram will cover any case where the friction force in pounds is 1/10 of the spring rate in lb per in. The resultant of the spring and damping forces is given by the resultant force line, which, of course, determines the acceleration of the vibrating mass.

Since the area under the resultant force line represents the value of the integral $\int FdX$, it, therefore, defines the work done. Since the vibrating mass starts from nest and again comes to rest after each half cycle, the positive work of acceleration must be equal to the negative work of deceleration, represented by the shaded areas above and below the base line. The same condition must be met on the return motion so

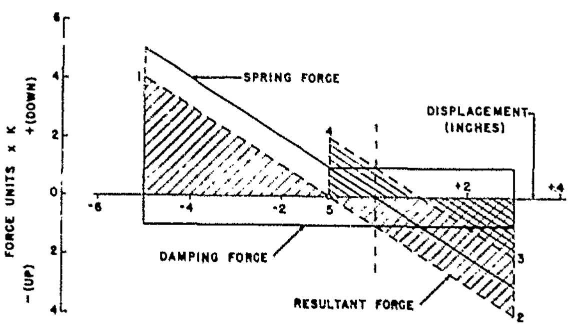


Figure 1 - Force and energy diagram of free vibration with coulomb damping
Conditions: Initial deflection (upward) = 0.5 in.
Friction (lb) = 0.10 x spring rate (K)

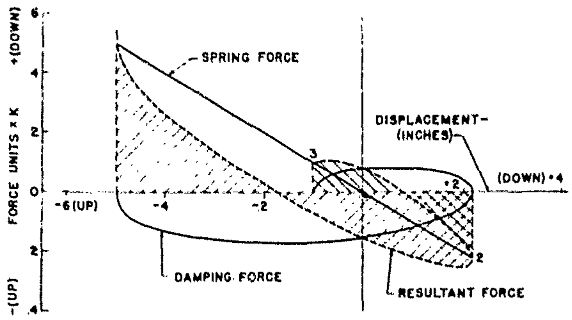


Figure 2 - Force diagram of free vibration with viscous damping
Conditions: Initial deflection (upward) = 0.5 in.
Damping constant = 25 percent critical
= 0.025 K lb/in./sec for W/K = 1

that at the end of one cycle the vibrating mass has come to rest at a displacement of 1/10 in. (points 4 - 5) or at a reduction in amplitude of 4/10 in. equal to four times the given ratio of friction to spring rate. It will be noted, however, for the case represented in this figure, that the resultant force at point 4 has become zero because the spring restoring force and the friction force are equal. Therefore, the free vibration is completely damped in one cycle for the conditions given.

A similar force diagram for viscous damping is given in Figure 2 for the same initial displacement and the same final displacement after one cycle. It will be noted that the damping force is now variable and the curve shown was developed point for point from the rationally derived equation of motion. The resultant force curve, as in the previous figure, meets the condition that the work during acceleration equals the work during deceleration, as indicated by the corresponding shaded areas. However, in contrast to the coulomb-damped vibration, the resultant force at the extreme displacements is necessarily identical with the spring restoring force since the damping force falls to zero at zero velocity.

Figure 3 shows the superimposed resultant curves for coulomb and viscous damping and brings out the following noteworthy points of difference in the characteristics:

(1) The maximum acceleration, which occurs at the beginning of the cycle, is lower with coulomb than with viscous damping. Moreover, with coulomb damping the rate of change of acceleration is uniform throughout the motion, corresponding to the constant slope of the resultant force curve.

(2) With viscous damping not only is the initial acceleration higher but the maximum rate of change of acceleration is also greater.

(3) The reduction in amplitude in the first cycle is greater with viscous damping but diminishes in the second half cycle to maintain the same ratio between successive extreme displacements. With friction damping the change in amplitude is uniform in each half of the cycle.

(4) Although the vibrating mass is completely damped after one cycle for the given conditions with friction damping, there is still a resultant acceleration acting upon the mass with viscous damping. Consequently, in the latter case, the mass will continue to vibrate although at diminishing amplitude.

(5) In the case of viscous damping, the maximum decelerating force acting on the mass is observed to occur at a point short of the maximum displacement. This is an important characteristic of viscous damping inasmuch as the damping force is necessarily out of phase with the displacement. This is the basic reason why the resultant force rather than the displacement amplitude is the really significant factor in viscous damping.

(6) The damping characteristic in free vibration also has an important bearing on the resonant excitation induced by discontinuous impulses, occurring at

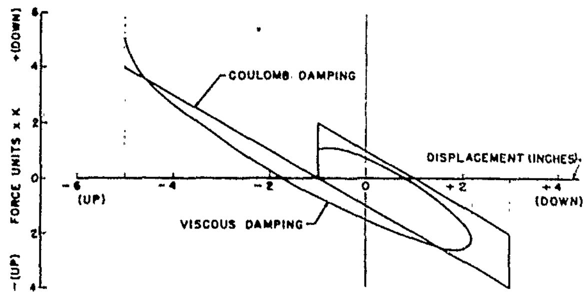


Figure 3 - Resultant force diagrams of damped free vibration - viscous vs. coulomb damping
Conditions: Static deflection = 1.00 in.
Initial deflection (up) = 0.50 in.
Coulomb friction = 0.10 K
Viscous damping constant = 25 percent critical

the natural frequency of the system. With coulomb damping the reduction in amplitude per cycle must be at least as great as the amplitude induced by the single impulse. Otherwise, the amplitude will build up on each successive cycle, theoretically without limit. This is obviously not the case with viscous damping.

An interesting sidelight on this comparison of viscous and coulomb damping is its interpretation in terms of human reaction in the case of vehicle-suspension damping. It has been found that, in the frequency range up to six cps (Reference 1), human tolerance of harmonic vibration is directly related to the maximum rate of change of acceleration, or "jerk." It might appear that the abrupt change in acceleration with coulomb damping at the extreme displacements, as a result of the reversal in the direction of the friction force, would contribute seriously to passenger discomfort. The fact is, however, that the disturbing effect of high jerk values is associated only with appreciable duration, as is indicated by the low frequency range of harmonic motion in which it is important. Since the high jerk rate has zero duration in the case of coulomb damping, it should have no discernible effect. On the other hand, the appreciable duration of the higher jerk rate with viscous damping should actually make the latter more uncomfortable. This conclusion is in keeping with actual observations in vehicles having suspension systems of one degree of freedom.

Steady-State Forced Vibration

The case of steady-state forced vibration presents a very different picture from that of free vibration and brings into much sharper contrast the comparative characteristics of coulomb and viscous damping.

Under steady-state forced vibration the transmitted forces through a simple vibrating system are well established both for the undamped condition and for viscous damping. However, as pointed out earlier, the case of coulomb damping does not lend itself readily to straightforward mathematical treatment. Nevertheless, Den Hartog (Reference 2) has worked out an ingenious approximation based on equivalent viscous damping which should be valid at least in evaluating the forced vibration amplitude. His analysis also brings out very clearly an important limitation of coulomb damping, namely, that it will not restrict the amplitude of resonant vibration unless the friction force is greater than $(\pi/4)P_o$, where P_o = the maximum amplitude of the driving force. This can be seen from the fact that the work done in 1/4 cycle by a harmonic driving force at resonance is equal to $(\pi/4)P_oX$, whereas the work done by the constant friction force, F is equal to FX. Consequently, if F is less than $(\pi/4)P_o$, the amplitude will increase on successive cycles theoretically to infinity. It is evident, then, that coulomb damping is not applicable to a system which is subjected to steady-state vibration at resonance unless the maximum value of the driving force is definitely limited and at a known level.

In connection with viscous damping characteristics, the treatment given by N. O. Myklestad (Reference 3) has been closely followed. This is exceptionally clear in differentiating between amplitude magnification and force transmissibility.

The comparison of viscous and friction damping effects in steady-state forced vibration has been made for the following specific conditions:

(1) Driving force proportional to the square of frequency $P = P_o \nu^2$

where

P_o = maximum driving force at resonance

$\nu = \omega/\omega_n$,

ω = driving force frequency, and

ω_n = natural frequency of system.

This condition was selected because it is characteristic of a vehicle transversing a fixed irregularity at variable speed; and of the unbalanced forces in all variable speed machines.

(2) Coulomb damping friction, $F = P_o$

(3) Viscous damping at 25 percent of the critical value. This gives substantially the same maximum force transmission as the specified coulomb damping in the resonant frequency range.

In Figure 4, the force-transmission characteristics for both conditions of damping as well as for the undamped system are given for the frequency range up to $\omega/\omega_n = 7$. The ordinate scale is in terms of the ratio of maximum force transmission (ft) to maximum driving force at resonance $(\omega/\omega_n = 1)$.

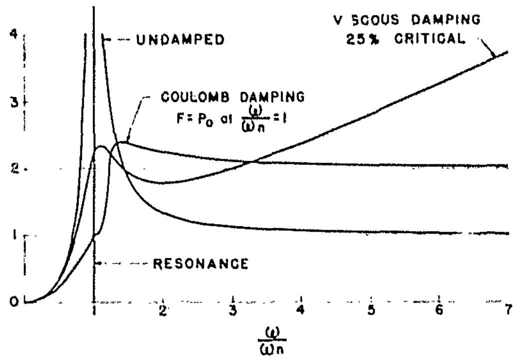


Figure 4 - Steady-state vibration force transmission at variable speed

It will be seen in this figure that the undamped condition, although capable of infinite amplitude at resonance, gives the lowest force transmission at high frequency ratios, approaching P_0 as a limit. With viscous damping at 25 percent of critical, the resonant amplitude is evidently well controlled but the force transmission at high frequencies increases continuously above a frequency ratio of 2 and begins to exceed the transmitted force at resonance at a frequency ratio of approximately 3.5. With coulomb damping at the specified level the maximum force transmission is substantially the same as with viscous damping near resonance but thereafter falls off continuously as the driving frequency increases, approaching a force transmission of $2 \times P_0$ as a limit. Thus, at the high frequency ratios coulomb damping adds to the force transmission only by the amount of the constant friction force.

It should be pointed out that, with friction equal to P_0, the friction force exceeds the maximum value of driving force, at all frequency ratios less than 1. Therefore, no motion of the vibrating mass is induced (see Figure 5) and the driving force is transmitted directly through the friction damper. Nevertheless, it will be seen that this force transmission is less than that obtained with either no damping or viscous damping in this frequency range. This follows from the fact that in either of the two latter cases the induced vibration introduces spring deflection forces which augment the resultant force transmission.

In contrast to the force transmission curves of Figure 4, Figure 5 shows the comparative displacement amplitudes of a steady-state vibration under the same set of conditions. Note that the ordinate scale is expressed in amplitude ratio with respect to the static deflection, P_0/K, corresponding to the maximum driving force, P_0, at resonance. As in Figure 4, the maximum harmonic driving force is assumed to vary as the square of the frequency ratio, so that

$$P = P_0(\omega/\omega_n)^2 .$$

It will be observed that, at high frequency ratios above 3, there is no difference in the displacement amplitude induced either with coulomb or viscous damping, or with no damping. Consequently, comparison on this basis gives not the slightest inkling of the greatly augmented force transmission that takes place through a viscous damper in this frequency range.

Figure 5 shows that the peak amplitude near the resonant frequency is 30 percent lower with the specified amount of coulomb damping than with 25 percent of critical viscous damping. This result must be qualified by the limitation, previously emphasized, that the damping friction must exceed $\pi/4$ times the maximum driving force at resonance to keep the amplitude from going out of control.

The following conclusions can be drawn from these results with regard to steady-state forced vibration of a system having a single degree of freedom.

(1) Where the operating conditions are confined to high frequency ratios, the undamped condition is the most desirable with regard to force transmission.

(2) Where it is necessary to control resonance, coulomb damping is very definitely to be preferred as long as the maximum driving force is known.

(3) Viscous damping is desirable only where operating conditions require resonance control under driving forces of unknown magnitude.

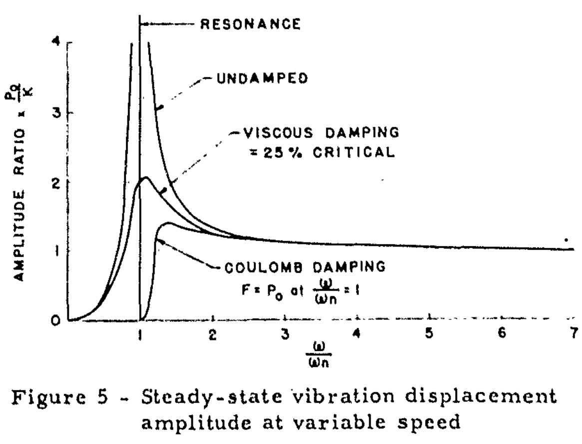


Figure 5 - Steady-state vibration displacement amplitude at variable speed

Transient Forced Vibration

The important practical condition of excitation by a single impulse is particularly difficult to evaluate by longhand, analytical methods. Consequently, the problem has been programmed on a digital computer and calculations are now in progress to determine the force transmission under each of the damping conditions investigated for steady-state vibration.

Preliminary results indicate that the relative magnitude of force transmission under transient excitation is very similar to the comparative results shown for the steady state at high ratios of natural vibration period to impulse duration. The comparable range of steady-state excitation would be at frequency ratios (ω/ω_n), greater than 3 to 1.

Impacts are commonly encountered in vehicle operation, corresponding to high amplitude impulses of very short duration. Under these conditions, the shock transmission with viscous damping, even when limited by flow-off valves, can build up to serious proportions.

Effect of Variable Mass with Fixed Damping

As pointed out previously, coulomb damping is independent of the vibrating mass and depends only on the relationship of friction force to spring rate. This characteristic makes it ideal for application to suspension systems which are subject to variable load. An excellent case in point is damping of railroad freight truck suspensions where the range of sprung load from full capacity to light car may be of the order of 5 to 1.

In the case of viscous damping, the relative damping factor depends upon the ratio of the absolute damping constant to the critical value. In a simple system the latter is defined as

$$C_{cr} = 2\sqrt{KM}.$$

Therefore, with a fixed spring rate and a viscous damper having a fixed damping constant, a variation in mass results in a change in the relative damping factor. This may be expressed as follows:

$$b = \text{damping factor} = \frac{C}{C_{cr}} = \frac{C}{2\sqrt{KM}}$$

If C and K are constants, then $\frac{b_1}{b_2} = \sqrt{\frac{M_2}{M_1}}$.

This relationship is shown graphically in Figure 6 by the curve designated "percent critical damping" and it will be seen that with a nominal damping factor of 25 percent at a relative mass of unity, the damping factor will increase to 35 percent when the mass is reduced one-half and will decrease to 18 percent when the mass is doubled. The corresponding amplitude ratios in free vibration, also shown in Figure 5 would be, respectively, 0.10 and 0.32 as compared with the value of 0.2, at the normal 25 percent damping factor.

Automobile Suspension Damping

Automobile suspensions, reduced to their simplest terms, are essentially systems of two degrees of freedom, considering each wheel as part of a separate system. This consists of a portion of the sprung mass M, the suspension spring of rate K, the wheel and associated unsprung mass m, and the tire having a spring rate k. Their relative magnitudes are commonly in the ratios

$$k/K = 7 \text{ to } 9 \text{ and } M/m = 6 \text{ to } 10.$$

The problem is to provide adequate damping of each of the two masses with a single interconnecting device. Since the natural vibration frequencies of the two masses are widely different, the damping requirements for each mass can be considered independently. Accordingly, the unsprung mass m, is acted upon by the tire and spring in parallel having a combined rate K + k, while the sprung mass can oscillate on the supporting spring of rate K.

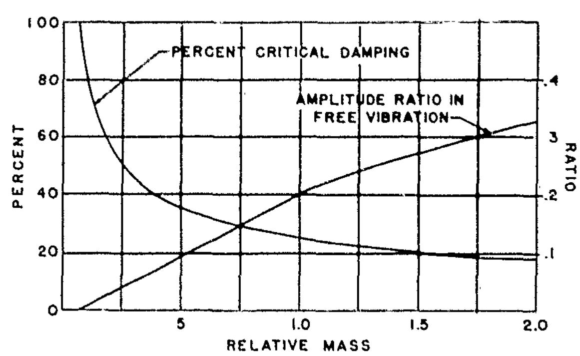


Figure 6 - Effect of variable mass with constant viscous damping—25 percent critical damping at relative mass = 1

If coulomb damping were to be used, the amplitude reduction per cycle in free vibration would be 4F/K+k for the wheel and 4F/K for the sprung mass. Since (K+k) is of the order of 8 to 10 times K, a given friction setting would produce only 1/8 to 1/10 as much damping effect on the unsprung mass as would be exerted on the sprung mass. This is quite evidently out of proportion to the possible amplitudes of motion.

On the other hand, the viscous damping factor is related to the product of spring rate and mass

$$(b = C/2\sqrt{KM})$$

so that, if C is a constant, corresponding to a common damper,

$$\frac{b_s}{b_u} = \frac{\sqrt{(K+k)\ m}}{\sqrt{KM}} = \sqrt{\frac{(K+k)\ m}{KM}}$$

where b_s = damping factor for sprung mass, percent of critical

b_u = damping factor for unsprung mass, percent of critical.

It is apparent then that the viscous damping constant will be of the same order for both masses, since the high combined spring rate and small unsprung mass have a product closely equivalent to that of the low spring rate and high sprung mass. Consequently, a viscous damper can be calibrated to provide the comparable damping requirements for both body motion and wheel hop.

ACTUAL CHARACTERISTICS OF HYDRAULIC AND FRICTION DAMPERS

As indicated in the introduction, dry friction dampers have fallen into bad repute largely because of the materials used for the friction surfaces. It is characteristic of practically all dry rubbing friction between metallic surfaces that the static and dynamic friction coefficients are widely different. In general, the starting or breakaway resistance is high and falls rapidly as the velocity increases. The obvious result is that the damper has a high impact transmission factor in relation to the energy absorbed during relative motion of the friction surfaces. Referring again to Figure 1, it is obvious that variable friction of this character, even in free vibration, would seriously increase the maximum acceleration for a given energy absorption.

Fortunately, it has been found that certain special nonmetallic molded materials, of the type employed in brake lining, in contact with metal surfaces approach very closely a uniform coefficient of friction, with the static and dynamic coefficients substantially identical. This means that the ideal coulomb type of damper can be made a reality. Furthermore, materials of this type have a much lower rate of wear and are insensitive to temperature, moisture conditions, etc.

The nature of variation in friction coefficient with velocity between any two materials can be strikingly demonstrated by a device illustrated in Figure 7. This consists of two drums geared together and driven counter to each other at the same speed, with the left-hand drum rotating in the clockwise direction. A bar placed across the two drums and fiven an initial velocity will obviously be forced to move at different relative velocities with respect to each of the two drums. If the coefficient of friction is independent of velocity the action of the bar will be stable. Once disturbed from its center position it will oscillate at the same or gradually diminishing amplitude. However, if the coefficient decreases with increasing velocity, the action of the bar will be unstable and a self-excited vibration will be induced until the bar either leaves the rolls or is otherwise restrained. It will be noted that the drums have two adjacent grooves so that the action of any two materials can be compared simultaneously. When a bar lined with the special composition material is placed on the drum side by side with an unlined steel bar, the stability of the lined bar and the instability of the unlined bar are immediately apparent. In fact, the unlined steel bar is self-exciting without external disturbance. It is of interest to note that this device is described by Timoshenko (Reference 4) as a means of determining experimentally the coefficient of friction, on the assumption that it remains constant. However, he does not mention the consequences of variability in the coefficient with velocity.

The mechanism of the action can be readily seen when it is considered that the friction driving force acting on the bar at each drum is a product of the weight reaction on the drum and the coefficient of friction. When the bar is displaced from its center position the friction force increases at the drum carrying the greater weight reaction and decreases at the drum carrying the smaller reaction. This force differential tends to restore the bar to its normal position, and it will be directly proportional to the displacement of the bar <u>only if the coefficient of friction is the same at both drums</u>. This will clearly be true when the bar is not moving.

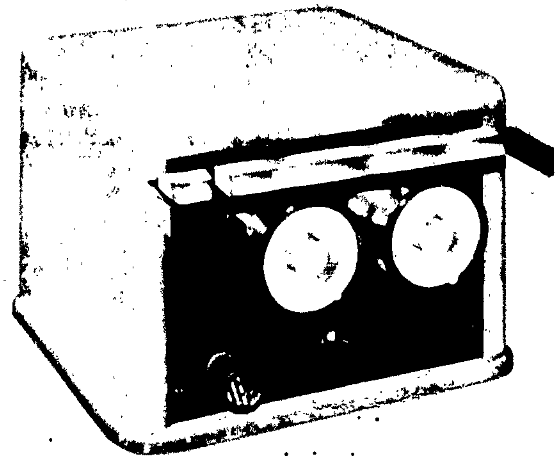


Figure 7 - Dynamic model demonstrating constant vs. variable friction coefficient with velocity

However, if the bar is in motion its relative velocity with respect to the two drums will be different, and, therefore, the linear relation between restoring force and displacement will hold only if the coefficient of friction is independent of velocity. Stated another way, the static and dynamic coefficients must be identical.

Now let us examine the consequences if the friction coefficient should decrease as the velocity increases, a condition characteristic of dry metal-to-metal friction. Since the relative velocity of the moving bar is greater with respect to the drum it is approaching, the coefficient of friction will be lower at that drum than at the one from which the bar is being displaced. This means that the force available at a given displacement to accelerate the bar will be greater than the force acting at the same displacement to decelerate the bar in the direction opposite from the center position. Since the work of deceleration must equal the work done during acceleration before the bar can be brought to rest, it follows that the final displacement will be greater than the initial displacement.

On the other hand, if the coefficient of friction is independent of velocity, the restoring forces will be equal for the same displacement in either direction and the initial and final displacement amplitudes will also be equal. Theoretically, therefore, the bar should continue to oscillate at a uniform amplitude once disturbed. However, with the special constant friction lining material it is found that the amplitude will diminish, gradually indicating that there is actually a slight increase in coefficient of friction with increase in relative velocity of motion.

At the left side of Figure 8 is shown an actual indicator card obtained with a constant friction snubber employing the special composition material operating against a soft steel barrel. This record was obtained on a machine which is positively driven with very nearly harmonic motion. It will be noted that the card is substantially rectangular and very nearly symmetrical about the base line, indicating practically constant friction throughout the stroke in both directions.

It is of interest to compare this record with the one on the right-hand side, obtained on the "Oriflow" type hydraulic damper, which probably comes closer than any other similar device to the true viscous damping characteristic. This comparison is brought out by the superimposed diagram which represents the theoretical variation in resistance in proportion to the velocity required to give the same energy absorption as that actually obtained. The superimposed dashed rectangular diagram represents the equivalent energy absorption with coulomb damping.

Figure 9 illustrates the insensitivity of the new constant-friction snubber to variations in velocity and temperature. The points plotted in (a) show the average friction obtained in tests made at different strokes and at different driven speeds. The scale of the abscissa is the product of stroke and rpm of the test machine and thus represents relative mean velocity. The ordinates show the average recorded frictional resistance in both directions. It will be noted that the overall variation from the average friction is of the order of ±4 percent over the entire relative velocity range from 20 to 300.

The lower curve shows that the total variation in frictional resistance between -60 and +160°F was confined to a range of 100 lb at a normal setting of about 1400 lb. In this connection it

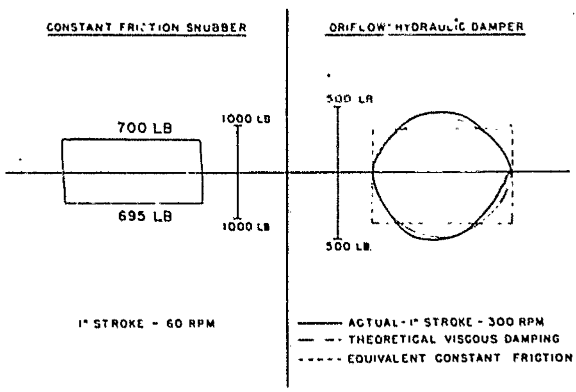
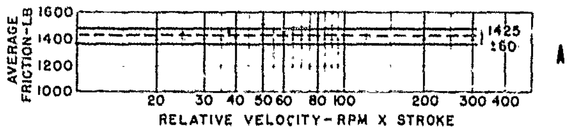


Figure 8 - Actual indicator cards



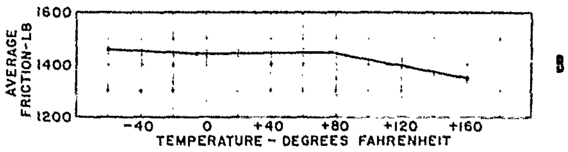


Figure 9 - Chrysler design constant friction snubber

(a) Average friction vs. velocity
- ⊙ 1/4-in. stroke
- × 1/2-in. stroke
- ⊡ 1-in. stroke
- △ 2-in. stroke

(b) Average friction vs. temperature 1-in. stroke at 60 rpm

will be noted that viscous dampers are necessarily sensitive to temperature by virtue of change in the fluid viscosity even with the relatively low viscosity index of the special fluids employed.

PRACTICAL APPLICATIONS OF CONSTANT FRICTION SNUBBER

The constant friction snubber was originally developed for application to the Chrysler design railroad freight truck and has been in commercial use on this equipment since 1948. The results have been highly successful both as to performance and life.

Extensive testing has proved that true coulomb damping produces a tremendous reduction in transmitted shock as compared with friction devices subject to high breakaway resistance and lacking precise control of the friction calibration.

Table 1 gives the comparative results on two railroad trucks having identical spring suspensions and loads where the only difference affecting the vertical shock level was the type of friction damping employed. The reference truck incorporated a built-in type of snubber using alloy cast iron shoes operated against hardened steel wear-plates. The comparison truck was the Chrysler design equipped with the constant friction snubber. Identical cars equipped with these trucks were operated in the same test train at speeds up to 90 mph over a course of approximately 120 miles. The vertical shocks were recorded on contact type accelerometers, each consisting of a bank of four elements calibrated to different acceleration intensities ranging from 25 percent to 75 percent of the truck acceleration. Table 1 shows the recorded counts of each shock intensity in two representative runs with different loadings and spring groups but with these conditions identical in both trucks in each test. It will be seen that the number of vertical shocks was far lower in the truck with constant friction snubbers. Reductions range from 87.5 percent to 72 percent at the light load and amounted to 96 percent in the lower accelerations with the heavy load. It will be noted that in the latter case no shocks were recorded with the constant friction snubber above the level of 35 percent, although a number of shocks above 50 percent were obtained with the reference trucks.

These results bring out another important factor with regard to suspensions, in general, namely, that the degree of shock protection provided by any suspension system is dependent basically on the ratio of load to spring rate, or static deflection of the spring. This is evident from the fundamental relationship between force, mass, and acceleration, $F = Ma$.

Since $F = KX$ for a given spring deflection from the static position (X)

and $M = W/g$,

$$a/g = KX/W$$

or acceleration is inversely proportional to the static deflection W/K for a given springing deflection.

Within the limited range of static deflection possible in freight truck suspensions, the shock transmission is especially sensitive to the load. For example, under the test conditions of Table 1 the light load had about 1 in. static deflection and the heavy load about 2 in. static deflection. The striking reduction in the number of recorded shocks with the heavier load, even with the inferior damping, testifies to the fundamental fact that there is no substitute for static deflection in a suspension system.

It is evident that higher static deflections can be obtained by reducing the suspension stiffness at a given load. This constitutes a potent argument for the use of special low rate spring groups

TABLE 1

Effect of Snubber Characteristic on Shock Level Railroad Freight Truck

No. of Accelerometer Counts at Stated Intensities in 120-Mile Run

3-11/16" Travel Springs - 145,000-Lb Capacity 73,000-Lb Rail Load Comparison of Shock Intensity				
	(25%)	(35%)	(50%)	(75%)
Built-in friction snubber / Reference truck (no. shocks recorded)	6187	1689	129	13
Self-contained constant friction snubber / Chrysler design truck (no. shocks recorded)	775	219	36	3
Percent shock reduction	87.5	87.0	72.0	77.0
3-11/16" Travel Springs - 169,000-Lb Capacity 145,000-Lb Rail Load Comparison of Shock Intensity				
	(25%)	(35%)	(50%)	(75%)
Built-in friction snubber / Reference truck (no. shocks recorded)	3165	835	8	0
Self-contained constant friction snubber / Chrysler design truck (no. shocks recorded)	120	30	0	0
Percent shock reduction	96.2	96.4	100	-

in freight cars that can be restricted to the handling of light loads of fragile commodities. This expedient has already been adopted by at least one important railroad which has equipped all of its fast merchandise freight cars with springs of reduced capacity and correspondingly lower rate in keeping with the light lading handled.

A more recent application of the constant friction snubber is its adoption by Army Ordnance for use on the latest medium M-48 tank, replacing the hydraulic dampers previously used. This decision was made after exhaustive tests by Ordnance development and field forces which conclusively established the superiority of the new friction unit. When both types of damper were calibrated for comparable control of hull oscillation, the following advantages were definitely established for the friction snubber:

1. Increased field service life in the ratio of at least four to one over the hydraulic.
2. Excellent stability of hull position after firing. This is vitally important, to avoid compensating for changes in inclination of the gun platform between rounds.
3. Reduction of noise, measured inside a light track vehicle, up to 12 db, with an average reduction of 8 to 10 db at three locations and at three different speeds.
4. Sizeable over-all decrease in hull vibration with maximum reduction of 75

percent in acceleration intensity at some frequencies and vehicle speeds.

The very pronounced reduction in noise and vibration is clearly in line with the predicted tendency of constant friction damping to minimize impact transmission. As shown in the analytical comparison with viscous damping, the latter tends to produce seriously augmented force transmission at high frequencies. This would tend to be exaggerated over the rough terrain encountered in cross-country tank operation.

Figure 10(a) shows a cutaway view of the constant friction snubber as applied to railroad cars, and 10(b) illustrates the similar unit adapted for tank installation. The internal construction is practically identical in both units; only the length of travel and external connections are different to suit the particular installation conditions. It will be seen in the figure that the friction elements consist of three segmental shoes with the special lining cyclewelded on the outer surfaces. The spring-loaded flat 45° wedge surfaces on the plunger head and free wedge ring engage mating surfaces on the shoes and thus apply uniform radial pressure to the shoes against the cylinder barrel. The lower wedge ring is necessarily free to slide on the plunger and thereby provides automatic take-up for wear. It will be noted that the shoes remain in fixed relation to the plunger and spring during operation of the snubber. Since the frictional resistance is approximately 60 percent of the spring pressure, the force required to compress the snubber cannot produce additional deflection of the spring.

These snubbers, developed by the Dynamics Research Department, Chrysler Corporation, are now being manufactured and sold by the Houdaille-Hershey Corporation, Detroit.

SUMMARY AND CONCLUSIONS

Analytical comparison of coulomb and viscous damping in simple suspension systems leads to the following conclusions:

1. In damping free vibration from a given initial amplitude, coulomb damping has the advantage of bringing the mass to rest in fewer cycles of oscillation. This comparison assumes less than critical viscous damping calibrated to give the same reduction in amplitude in the first cycle. Where the system is excited by periodic impulses at the resonant frequency or some lower harmonic, coulomb damping will be adequate only if the oscillation can be completely damped between successive impulses.

2. In damping steady-state forced vibration, coulomb damping is definitely superior to viscous damping because of its very much lower force transmission at high frequency ratios.

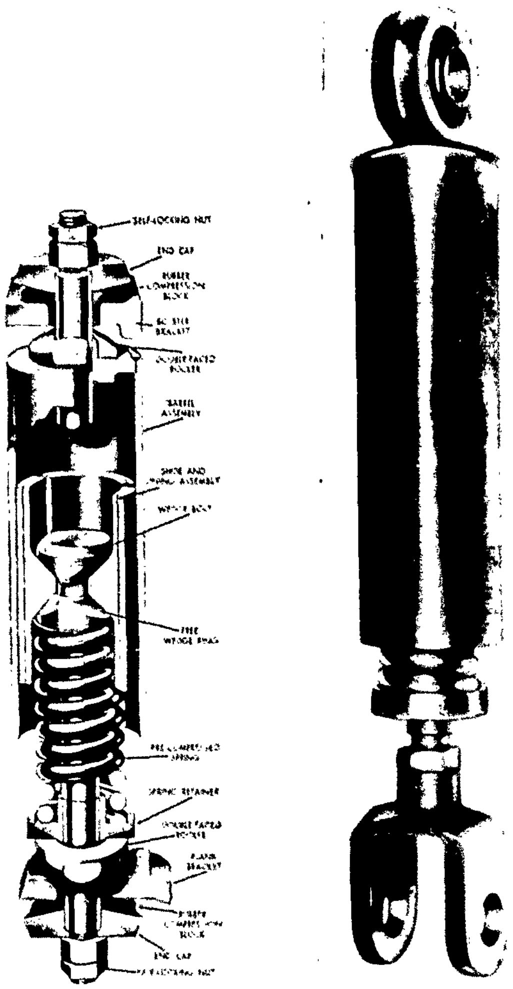
   However, the coulomb friction must exceed 78.5 percent of the maximum driving force at resonance in order to control the vibration amplitude. Therefore, if resonance occurs in the operating speed range, coulomb damping is applicable only if the maximum driving force is a known quantity.

3. Coulomb damping attains its maximum advantage over viscous damping when the system is subjected to transient disturbances, or impacts, of short duration in relation to the natural period of vibration. This is the result of its inherently limited force transmission as compared with the unlimited force transmission possible through a viscous damper at high velocities.

4. Coulomb damping possesses the additional advantage of being independent of the mass of the system and related only to the spring rate. This is of great practical value in such vehicles as freight cars where the sprung load commonly varies over a range of five to one between the light and fully loaded car.

5. Preliminary analysis of coulomb damping as applied to automobile suspension systems indicates that a fixed amount of friction is not inherently well adapted to dampen both the sprung and unsprung masses. On the other hand, a single viscous damper can exert a comparable degree of damping on both masses.

The Chrysler design constant friction snubber achieves true coulomb damping by the use of friction materials which are unique in having the same static and dynamic coefficients. Test results are given for production units as successfully applied to railroad freight trucks and Army track vehicles. These may be summarized as follows:



A. B

Figure 9 - Chrysler design constant friction

1. The average friction is shown to be insensitive to velocity over a wide range of stroke and driven speed; and to change very little with working temperatures varying from -60° to +160°F.

2. In simultaneous road tests of two types of freight truck having identical springs and loads, the one equipped with constant friction snubbers consistently recorded only a small fraction of the vertical shocks obtained with the comparison trucks which incorporated metal to metal friction damping.

3. Exhaustive tests by Army Ordnance development and field forces on tanks and other track vehicles have proved that the constant friction snubber has an average effective life at least four times that of hydraulic dampers. With both types of dampers calibrated to give comparable control of hull oscillation, the friction units markedly reduced the vibration and noise levels and provided a more positively positioned gun platform. On the basis of these results the Chrysler design snubber has now been specified as mandatory equipment on all M-48 tanks.

ACKNOWLEDGMENT

The invaluable assistance of Mr. F. F. Timpner, project engineer of the Chrysler Dynamics Research Department, in the mathematical analysis, planning, and carrying out the digital machine calibrations, are gratefully acknowledged.

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DISCUSSION

A. M. Weston, General Electric: I would like to know if there are materials available that will give coulomb friction at relatively high metal temperatures on the order of 1000 or 1200 degrees Fahrenheit and, if so, to what general class these materials belong.

Janeway: I don't know about the temperature level, but the material that we use is a special type of molded brake lining; which in itself is temperature resistant and, of course, is designed for operation at high temperature. However, it cannot be used at a higher temperature than burns paint off the unit. I would say there is a very good chance that this material would meet your requirements. I don't know what strength you need and so forth. What we use is a special lining running on soft steel.

* * *

TYPE APPROVAL OF GUIDED MISSILE CONTAINER SHOCK ISOLATORS

Kauko Kuoppa-Maki, NBS, Corona

An approximation of parameters influencing the reliability characteristics of shock isolators is represented as a function of design features and expected changes during storage and fatigue aging. The suggested type approval is based on the performance characteristics at the end of an expected service life, and at environmental extremes.

INTRODUCTION

The Missile Evaluation Section of the Missile Development Division at the National Bureau of Standards, is responsible for the preparation of inspection instructions for Terrier-container shock isolators. This analysis is based on current Terrier specifications, on recent Shock and Vibration Bulletins and other available literature, and on studies performed jointly by the Dynamic Group, Numerical Analysis Group, and Preservations Group of the NBS Missile Evaluation Section.

The problems of shock isolation, although applied here to Terrier missile containers, are basically the same for all missiles. In Terrier terminology, the inspections are divided into four phases. Type approval, the basic inspection, determines the conformity of the component performance with the requirements outlined in the missile specifications. Type approval gives the fundamental information needed in routine checkups such as the quality control type tests of the production phase, the final acceptance inspections performed in the delivery phase, and the surveillance inspections to determine component reliability during transit and storage.

The characteristics of shock isolator performance can be given by three main parameters: (1) The required reliability of shock isolation; (2) the tolerated maximum accelerations expected for the packaged component; and (3) the expected load in the form of shock, vibration, and storage aging during transit and storage at extreme conditions.

THE REQUIRED RELIABILITY LEVEL

The basic requirement in shock isolation is to maintain a specified level of reliability.

The design aspect of reliability characteristics is expressed for a system of guided missile components with the formula (Reference 1)

$$R_S = \left[1 - (1 - r)^m\right]^n. \quad (1)$$

In this formula, n represents the number of series-connected operating bands acting in series. Failure of one band causes the system to fail. m represents the number of units in one band, acting in parallel. If one unit operates, the system will continue to operate. r is the unit reliability.

In regard to reliability of shock mount systems this equation is written

$$R_S = (r^m)^n. \quad (2)$$

In Equation (2), the symbols n and r are as in Equation (1), but in this equation m represents the number of shock isolator units in parallel, designed in such a way that one failure causes the isolator band to fail.

The reliability of a shock isolation system, however, can be improved by increasing the number of shock isolator units which can fail without the system failing.

To make the reliability analysis possible, Equation (2) is rewritten in the form

$$R_S = \prod_{i=1}^{n} R_i \qquad (3)$$

where R_S represents the system reliability and R_i, the reliability of the i^{th} shock isolation band. The symbol i goes from 1 to n.

The isolation band reliability

$$R_i = \sum_{\lambda=0}^{k_i-1} \left|\begin{matrix} m_i \\ \lambda \end{matrix}\right| r_i^{m_i-\lambda} f_i^{\lambda} + a_i r_i^{m_i-k_i} f_i^{k_i} \qquad (4)$$

contains two terms. The first term comprises the first k_i terms of the binomial expansion of $(r_i + f_i)^{m_i}$, in which λ represents the number of units of the i^{th} band which fail in the various possible alternatives, and k_i represents the number of units in the i^{th} band which will cause the system to fail; $\left|\begin{matrix} m_i \\ \lambda \end{matrix}\right|$ is the number of ways in which λ units can fail. In the second term, the coefficient a_i represents the number of ways in which k_i units can fail without the system failing; this depends on certain system restrictions explained in Table 1.

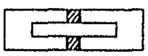
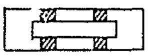
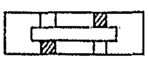
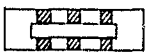
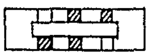
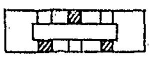
The above schematic example on how design aspects can effect reliability is presented to show that an intelligent approach to type approval of shock isolators needs a definition of the reliability requirements which the system shall meet. Reliability level required is a basic design criteria for shock isolators.

TABLE 1
Reliability of a Shock Isolator System

$$R_S = \prod_{i=1}^{n} \left[\sum_{\lambda}^{k_i-1} \left|\begin{matrix} m_i \\ \lambda \end{matrix}\right| r_i^{\,m_i-\lambda} f_i^{\lambda} + a_i r_i^{\,m_i-k_i} f_i^{k_i} \right]$$

Conditions:

Number of isolator bands	$n = 1$
Number of the band in question	$i = 1$
Reliability of a isolator unit	$r_i = 0.9$
Probability of a unit failure	$f_i = 0.1$
Number of isolator units in parallel, m_i	
Number of unit failures needed for the isolator band to fail, k_i	

m_i	k_i	System Restrictions	Formulas		System	
2	1	None; i.e., if any unit fails, the ith band fails	$R_S = (0.9)^2$	= 0.81		1a
4	1	None	$R_S = (0.9)^4$	= 0.66		2a
4	2	For the i^{th} band not to fail, the failed units must be oriented diagonally, $a_i = 2$	$R_S = (0.9)^4 + 4(0.9)^3(0.1) + 2(0.9)^2(0.1)^2$	= 0.96		2b
6	1	None	$R_S = (0.9)^6$	= 0.53		3a
6	2	For the i^{th} band not to fail, the failed units must be diagonally opposite or be a center pair, $a_i = 3$	$R_S = (0.9)^6 + 6(0.9)^5(0.1) + 3(0.9)^4(0.1)^2$	= 0.905		3b
6	3	For the i^{th} band not to fail, the failed units must be located as each second one, $a_i = 2$	$R_S = (0.9)^6 + 6(0.9)^5(0.1) + 15(0.9)^4(0.1)^2 + 2(0.9)^3(0.1)^3$	= 0.986		3c

NOTE: The figures on the right indicate the greatest number of isolators that can fail ( ) without the system failing. The shaded isolators ( ) indicate the isolators that have not failed.

Reliability as an aspect of economy is shown in Figure 1. It is assumed in this illustration that the syster reliability has to meet a reliability level of 0.76 as a minimum requirement. Two shock-isolator systems, (a) and (b), have been designed to meet this requirement. System (a) is inexpensive in construction but requires special care in handling. System (b) has a more elaborate construction, but gives satisfactory reliability with normal handling methods.

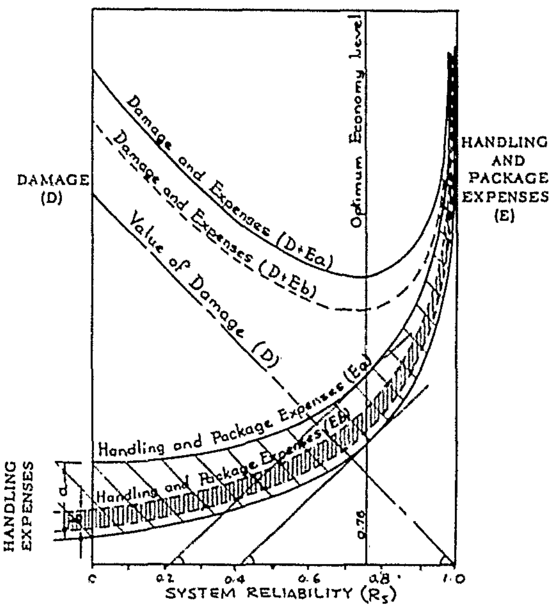


Figure 1 - A schematic representation of the economic aspects of reliability

The maximum economy is reached on a level of reliability where the curves Ea or Eb of handling and packaging expenses have the same slope in absolute value as the line of decreasing value of transit damage D. This is expressed by

$$\frac{dE_a}{dr} = \frac{dE_b}{dr} = D\Big|_{R_S = 0} \qquad (5)$$

The optimum economy level (vertical line on 0.76 reliability in Figure 1) illustrates the well-known limit between underdesign and overdesign in packaging. The reliability of the container can be increased by improvements incorporating additional expenses. Above the optimum economy level, however, the additions in expenses are greater than the savings in decreased damage.

In shock isolator type approval, the 1.0 limit of reliability cannot be considered a realistic figure. It is possible to improve shock isolation in order to get closer to a 100 percent reliability, but the expenses increase. It can be stated that the expenses approach infinity when the reliability level approaches 1.0.

THE TOLERATED MAXIMUM ACCELERATIONS

Transient Shock

The maximum accelerations that the missile components can tolerate are specified in terms of vibration load (Reference 6) and flight accelerations (Reference 2). The trend is towards increased ruggedness, and the present missile component test requirement of 10 and 40 g's transverse and longitudinal accelerations, respectively, is suggested to be increased to 20 and 60 g's (Reference 3), as per Table 2 below.

TABLE 2

Definition	Direction of Impact	
	Transverse	Longitudinal
Official range (Reference 2, paragraph 4.2.7.2)		
acceleration	10 g's	40 g's
build-up time	-	0.03 sec
duration time	40 sec	0.03 sec
Suggested range (Reference 3)		
acceleration	20 g's	60 g's
build-up time	-	0.03 sec
duration time	40 sec	0.03 sec

The duration time in the above table refers, in the longitudinal direction, to impact during boost phase. In the transverse direction it refers to simulation of steady flight accelerations. Of these two, the transverse acceleration can be expected to be the critical one during handling and transit. However, the velocity change in the frequency spectrum during handling and transit is of a different nature. It has been shown for mild steel (Reference 12) that a steady state of acceleration is more severe than an impact representing the same number of g-units. This can be true for most missile components. It should be noted though, that if the same impact is repeated, the cumulative load can become more severely damaging than the corresponding steady-state acceleration. Because of this difference, the acceleration limits specified for

component testing, Table 2, cannot be accepted as limits of accelerations during transit. It is advisable at the present state to go towards over-design in the shock isolation and specify less than 20 and 60 g's or 10 and 40 g's for the maximum tolerance of the packaged component—at least until the time of duration and the expected frequency occurrence of velocity shocks on packaged components have been defined. Niles suggest (Reference 4) that 7 g's be the limit in the transverse direction and 30 g's in the longitudinal direction.

Steady-State Vibration

The specifications (Reference 6, paragraph 4.5.1) require that the shock isolators maintain the missile vibration below 5 g's within 10 to 300 cycles when the container is subject to a steady-state vibration corresponding to 3.5 g's in frequency of resonance of the packaged mass-spring system.

It is understood that the vibration problem can be solved by an independent vibration isolator constructed to act in series with the shock-isolation system. This makes it possible to block the shock isolators for the steady-state transit period if the amplification of the shock isolators appears to become critical in their low-frequency range (see Figure 2).

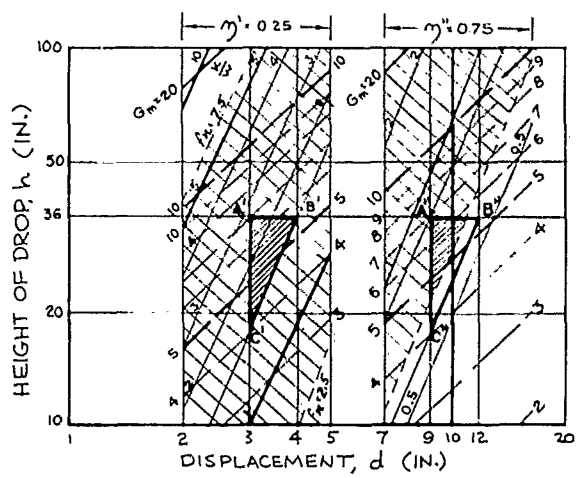


Figure 2 - Shock isolator characteristics

THE EXPECTED LOAD IN FORM OF SHOCK, VIBRATION AND STORAGE AGING

The expected conditions during handling transit, and storage are described for Terrier containers in NAVORD specifications, (References 5 and 6), and in suggested Convair revisions (Reference 7). They call for the following:

Storage Age

Five years' storage in dehydrated containers (Reference 6, paragraph 3.1.3.1), has been specified as minimum for the missile components.

Temperature Extremes

The temperature of the shock isolator is a function of the temperature of the environment and of the design features of the isolation system.

The environmental temperature has several specified extremes:

1. -65°F to +160°F on open dock or during shipment (Reference 5, paragraph 3.1.7.1.1),

2. -85°F to +160°F during flight or air transit (Reference 5, paragraph 3.1.7.1.1).

The following additional temperature limits have been given:

1. 0°F to +120°F within the container on board ship (Reference 2, paragraph 3.1.7.1.1)

2.2. -20°F minimum for the booster and sustainer grain in flight readiness (as per information from Hercules Powder Co.).

The operating temperature of the shock isolator's represents a basic design criterion. It is felt that the influence of the prevailing conditions on the operating temperature of the shock isolators needs further attention.

The specified maximum temperature (+160°F) corresponds to the temperature inside the container when solar radiation is not considered. The effect of solar radiation and the rise in temperature caused by hysteresis within the shock isolator during steady-state vibration in transit, should be considered. Niles (Reference 4) suggests the requirement that the shock isolators should show good performance in temperatures up to +350°F for short-time duration.

Since the lowest recommended temperature for booster and sustainer is -20°F, it would be advisable to consider incorporation of auxiliary heat and isolation against heat loss to maintain

the -20°F temperature limit inside the containers of other missile components as well.

In accordance with the above analysis, -10°F to +350 F° could be anticipated as the service extremes of the surface temperatures of shock isolators.

Vibration Load

The cumulative fatigue load during transit is expected to reach the maximum during a cross-country rail shipment. Vitro Corporation suggests the following method for simulation of this fatigue load (Reference 7, pp. 6,9):

"Vibrate the loaded container at the resonant frequency, using a force of 2-1/2 ± 1/2 g's maximum, for 10 hr. . .using a conditioning temperature of 160°F. . ."

Vitro Corporation states (Reference 7, p. 18) that the above simulation of the expected cumulative fatigue load is tentative and suggests that the crated missile as well as crated individual missile-components actually be shipped by truck, rail, aircraft, and marine transportation in order to compare the severeness of the simulated fatigue load with the true shipping conditions.

Velocity Shock

The maximum impact load expected has been specified to correspond to a drop of one end of the container on a concrete base from a 36-in. height (Reference 6, paragraph 4.5.2 and Reference 8, paragraph 4.3.2.1). It can be assumed that the final cause of the critical failure in most cases is a velocity shock. The parameters influencing isolation of a velocity shock are therefore of primary importance in type approval.

Figure 2 describes an analytical approach to the determination for type approval purposes of the characteristics of the shock-isolation parameters. The space for displacement needed for a given height of drop is shown as a function of aging and temperature change on a shock isolator with linear elasticity, considering inelastic impact. Viscosity is considered negligible. The two distinct groups of parameters represent two extreme cases. The single-prime group at the left represents the optimum case with least space requirement. The double-prime group at the right represents the opposite extreme, the greatest space requirement to be expected.

Each point on Figure 2, within a limited area, represents definite values of the following functions:

h_d = height of drop (in.) or more precisely, the vertical distance of travel of suspension point of the shock isolator from start of the fall to the point of impact. The subscript d is explained in connection with Equation (9).

d = displacement of the component within container (in.)

G_m = (a_m/g), the maximum deceleration a_m in g-units.

(k_c/W) = specific spring rate (lb/in. displacement/lb suspended).

The subscript c is explained in connection with Equation (11).

$\begin{pmatrix} k_c = \text{spring constant (lb/in. deflection)} \\ W = \text{weight of suspended component} \end{pmatrix}$

f_n = natural frequency (cps) of the suspended mass-spring system.

The following comments also apply to Figure 2:

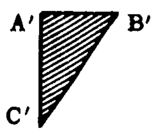
A′ and A″ = space needed at -20°F with an aged isolator

B′ and B″ = space needed at +350°F with a factory-fresh isolator

C′ and C″ = height of drop at +350°F within the space reserved for displacement at -20°F

η' = energy transmission coefficient 0.25

η'' = energy transmission coefficient 0.75. (η = 1.0 stands for vibration with one degree of freedom and without damping)



is the "d/h" area for a shock isolator system with η'. (The area A″ B″ C″ calls for a system with η''.)


is the critical frequency area in railroad transportation, $2.5 < f_n < 7.5$.

The G_m, k_c/W, and f_n parameters in Figure 2 follow equations:

$$G_m = \frac{h_d}{d} - 1 \tag{6}$$

$$k_c/W = \frac{1}{d}\left(\frac{h_d}{d} - 2\right) \tag{7}$$

$$f_n = \frac{g}{2\pi}\,\frac{G_m^2 - 1}{h_d}\,. \tag{8}$$

These equations are based on Mindlin's methods of analysis (Reference 9) and further derivations of McNamee (Reference 10) for cases where displacement d is not small with respect to height of drop h.

The preparing of a shock isolation evaluation tool, as illustrated in Figure 2, is based on two hypotheses:

Hypothesis I; The design characteristics c_d can be expressed, without too great an error, as a coefficient influencing the effect of the height of drop h, when the other variables are kept constant,

$$h_d = c_d\, h_o \tag{9}$$

where

$$c_d = \eta e \tag{10}$$

h_o = original height of drop (in.)
h_d = design height of drop (in.)
e = elasticity coefficient is obtained by analytic derivation from the static load displacement characteristics;
e = 1 for optimum spring in preloaded linear elasticity with k = 0 (called hyperbolic tangent elasticity in Mindlin's derivation);
e = 2 for a linear spring with any k values.

Further e - values are given in Table 1 of reference 10.

η = efficiency coefficient of energy transmission. It is an empiric function of the number of degrees of freedom and the various kinds of friction losses encountered in changing the potential energy of the falling mass to the potential energy of the displaced spring. The characteristics of the efficiency coefficient are not known in detail.
η' = 0.25 corresponds approximately to the least possible amount of potential energy transmitted to the displaced spring (the single-prime symbols in Figure 2);
η'' = 0.75 represents the opposite extreme, (the double-prime symbols in Figure 2).

Hypothesis II: The conditional characteristic (c_c) representing changes expected in the stiffness of the spring (k) can be expressed without too great an error, with the following formulae:

$$k_c = c_c k_o \tag{11}$$

$$c_c = c_t c_a c_i \tag{12}$$

where
k_o = original spring rate of a factory-fresh shock isolator at the highest temperature extreme.
k_c = conditional spring rate of an aged shock isolator at the lowest temperature extreme.
(c_t) = temperature change characteristic; it indicates the change in isolator stiffness caused by a change in temperature. The values of this characteristic are easily determined for short duration of temperature change, but as Crede indicates (Reference 11, p. 240), the influence of time on this characteristic is not clearly defined. For a temperature change from +350°F to -20°F, a value c_t = 1.6 for rubber compounds is used as a rough approximation in this analysis.
(c_a) = aging characteristic; it indicates the change in spring constant (k) caused by fatigue and storage aging in the extreme conditions. Unit value c_a = 1 represents conditions where such change is not expected. The nature of this characteristic is not known. For shock isolators made of rubber compounds, an approximate value c_a = 1.25 for an aging corresponding to 5 years of storage in environmental extreme is used in this analysis.
(c_i) = impact characteristic. Unit value c_i = 1 is tied to conditions corresponding to an inelastic impact as experienced in drops on concrete. For elastic impact, $c_i > 1$ because the elasticity of impact can be considered to represent a spring acting in series with the shock isolator inside the container.

The inelastic impact is a good yardstick in evaluation of shock isolation. It is felt, however, that because of the difficulty in repeating the impact conditions in the drops required, the number of drops specified by the present specifications (Reference 8) is not large enough for obtaining a satisfactory confidence level for the mean

value. For this reason it might offer advantages in economy to change from inelastic impact to test conditions with controlled impact. A number of test procedures with controlled elasticity of impact have been developed (Reference 13).

In Figure 2 it is presumed that impact is inelastic ($c_i = 1.0$) and that the shock isolators are factory-fresh vulcanized-rubber shear blocks with a specific spring rate $k_o/W = 1.3$ lb/in. displacement/lb suspended at +350°F. It is anticipated that the stiffness will increase when aging affects the shear blocks ($c_a = 1.25$) and when the temperature drops to -20°F (c_t)= 1.6). According to this approximation the corresponding conditional k_c/W value at -20°F in the $\eta' = 0.25$ area is

$$k_c/W_{-20^\circ} = c_i \cdot c_a \cdot c_t \cdot k_o/W_{+350^\circ}$$

$$= 1.0 \cdot 1.25 \cdot 1.6 \cdot 1.3 = 2.6 \text{ lb/in./lb} \quad (10)$$

and correspondingly in the $\eta'' = 0.75$ area

$$k_c/W_{-20^\circ} = 1.0 \cdot 1.25 \cdot 1.6 \cdot 0.44 = 0.88 \text{ lb/in/lb.}$$

The location of these stiffnesses can be seen between corresponding k_c/W lines in Figure 2.

Point A′ in the $\eta' = 0.25$ group in Figure 2 represents the characteristics of this shock isolator at -20°F at the end of the service life (k_c/W = 2.6 lb/in./lb, h = 36 in., and d = 3 in.). When factory-fresh and at +350°F, this same isolator with a softer spring constant (k_o/W = 1.3 lb/in./lb) requires a larger space for displacement (d = 4.3 in.) when dropped from the same height (h = 36 in.).

If space for this 4.3-in. displacement is not reserved within the container the conditions could also be described in the following way: The system is good for drops up to 18 in. high when the factory-fresh shock isolator has a +350°F temperature (point C′ in Figure 2). The height of drop can be increased to 36 in. when the shock isolator becomes fatigue aged and is in a -20°F temperature (point A′ in Figure 2).

The triangle A′ B′ C′ represents the effect of aging and temperature change on space needed for displacement in the isolation system with $\eta' = 0.25$. The same holds for the corresponding triangle A″ B″ C″ in the η'' group of parameters.

According to this approximation the distance A′ A″ represents the range of original space requirement, $d_o' = 3.0$ and $d_o'' = 9.0$. The distance B′B″ represents the range of conditional space requirement, $d_c' = 4.9$ and $d_c'' = 12.8$.

In type approval, the empiric mean of efficiency coefficient η will be determined by drop testing, and corresponding G_m, k_c/W, and f_n lines will be drawn on the h/d log/log graph. These determine the characteristics of the analyzed shock isolator in the form of space ABC. If for example the mean efficiency coefficient η has been determined to be 0.59, then the corresponding point A, the original space for displacement, requires

$$d_o \simeq 5 \text{ in.}$$

and the point B, the conditional space for displacement, will require

$$d_c \simeq 7 \text{ in.}$$

considering the other parameters have remained the same.

It is to be presumed that changes and modifications are needed on the method of type approval, using the curves of Figure 2 as evaluation tools. This is to be expected, especially when more material is available on shock isolators with high resiliency influencing both the design characteristic (c_d) and conditional characteristic (c_c). In the present analysis, however, the method expressed in Figure 2 is presented mainly for illustrating the relative effect of the various parameters in expected conditional extremes.

To illustrate the expected changes in the various shock-isolator systems, corresponding sets of parameters can be drawn also for a preloaded linear elasticity, a cubic elasticity, or a tangent elasticity (Reference 10), in case any one of these appear to give a closer fit to the empiric values obtained in the type approval tests.

A Statistical Approach

The foregoing analysis of shock-isolation parameters gives a preliminary approximation of various functions to be determined by specific laboratory tests.

The conformance of the laboratory tests with corresponding functions in actual service, can be determined by two methods: (a) controlling shipments with recording instruments and (b) collecting case histories of "service difficulties" experienced during handling and transit.

Material collected from recorded shipments and case histories in general, can be used in analyzing the following phases of the problem: First, the probability of appearance of shock load peaks of critical nature during specified shipping conditions, second the integration of the cumulative fatigue load of specified transit conditions, and third, the number of shock isolator failures per transit unit, classified in accordance to severeness of failure and type of transit.

The use of statistical tools will become a further improvement when coordination in specifications and test procedures makes it possible to reflect the total of missile handling experience, all missile types included, within the same population for statistical analysis.

CONCLUSIONS

The above approximations lead to the conclusion that it would be desirable for the specifications on shock-isolation requirements to determine the characteristics of shock isolators based on a minimum of acceptable reliability in expected extreme conditions at the end of the required service life.

The material originating from type-approval tests is needed in the first place for confirming the conformance of the shock isolator characteristics with the specification requirements. The same material, however, furnishes the basic foundation for designing the other test procedures, especially for the surveillance testing intended to control the reliability level of shock isolation during the service life. Whether a change in shock-isolator characteristics during the service life is anticipated or not, the type approval is expected to be focused to illustrate the functions of shock and vibration isolation at the end of the service life.

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DISCUSSION

Lt. Orensteen, WADC: I would like to ask if the chart which you have drawn showing the relationship between cost and reliability also includes the cost of failures? In other words, do your curves include the cost of failures, or is that something that has yet to be added to the total cost curve to determine the reliability with which it will operate?

Kuoppa-Maki: That is a good question. You will remember the straight line which gave the optimum limits where the cost came down and went up again; this line gave the minimum cost. This represents the total expense, both of failure and packaging.

It simply puts in picture form the idea of overdesign. If we go higher up in reliability there is a certain line where we exceed the reasonable expenses. We often speak about a hundred percent reliability; but if we try to get one hundred percent reliability, the expenses of packaging will go higher up and mathematically speaking, reach infinity when we obtain one hundred percent reliability.

There is an optimum line and this research of shock isolation should find out the optimum line for each particular case.

Orensteen: I notice from the shape of the cost curve that it descends more slowly to the optimum point than it rises after the optimum reliability. Can one say which would be cheaper if one had the choice of either overdesigning or underdesigning ten percent? You notice that as you exceed this optimum reliability your costs rise rather rapidly, whereas if you underdesign in reliability your costs don't rise rapidly as you reduce your reliability.

Is it more economical to underdesign than overdesign?

Kuoppa-Maki: I think you are right. In this particular case, underdesign is more economical than overdesign.

R. E. Blake, NRL: Have you had any experience in case where an increase in the number of degrees of freedom causes larger deflection than the single degree of freedom?

Imelda McNamee, NBS Corona: Our energy transmission coefficient η should be considered only as a preliminary simplification of the complex phenomenon of various degrees of freedom.

In the simple case considered in the paper, η is either equal to one (linear motion only) or less than one (more than one degree of freedom). This means that an increase in number of degrees of freedom always gives a decrease in deflection.

Blake: We have learned that if torsion is present in a shock isolator, a larger space is sometimes needed for displacement than if the motion is only linear.

McNamee: For the type of isolator I mentioned, the addition of torsional motion decreases the deflection. However, consider a shock isolator in which the springs are arranged in such a manner that kinetic energy can oscillate between them when there is torsional motion. In this case, the center of gravity of the mass being isolated moves a small amount compared to the displacement of the extreme parts of the component supported by the springs. In such case, the addition of rotational motion could increase the space needed for displacement, as you mentioned, and would be larger than one.

In fact, η is merely a parameter which modifies (increases or decreases) the formula for displacement d due to motion in more than one degree of freedom.

* * *

DISPLACEMENT NEEDED IN SHOCK ISOLATION

Imelda McNamee, NBS, Corona

An analytical approach is made to the problem of finding the displacement needed in a shock isolator when the mass to be isolated is drop-tested from a specified height and maximum deceleration results. The natural frequency of the shock isolator is analyzed as a function of the same height of drop and maximum deceleration.

INTRODUCTION

This analysis follows the lines of R. D. Mindlin's method (Reference 1). Relatively large displacements will be considered.

The shock isolator system will be represented simply as a mass-spring system which is to be dropped onto a perfectly rigid surface from a height $\underline{h}$, Figure 1. The displacement of the mass m due to the impact force at loading is represented by $\underline{x}$, Figure 2.

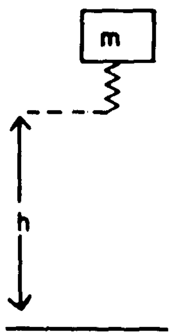


Figure 1

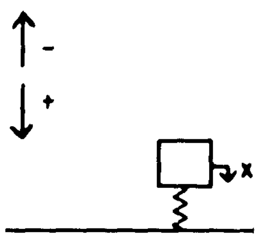


Figure 2

Let g = acceleration due to gravity,

G_m = maximum deceleration of $\underline{m}$ in terms of number of g's,

d = maximum displacement of mass $\underline{m}$ from its position at the moment it lands,

f_n = natural frequency of mass-spring system.

The units used are inches, seconds, and pounds. Force is in poundals and spring constants in poundals per inch deflection.

In Table 1, $\underline{d}$ and f_n are expressed as functions of $\underline{h}$ and G_m for various types of springs.

ANALYSIS

The expressions for d in Table 1 may be written in the general form:

$$d = e\,\frac{h}{G}, \tag{1}$$

where e = the elasticity parameter (depending on the type of spring),

$G \simeq G_m$ (defined separately for each type of spring).

In the following analysis, values of $\underline{e}$ will be derived for the several different types of springs.

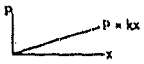
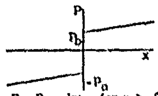
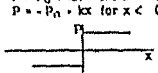
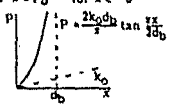
For any type of spring,

$$ma = -P + mg \tag{2}$$

where a = acceleration of m (thus $a_{max} = -gG_m$)

P = force on the spring.

TABLE 1

Type of Spring	Static Load Displacement Curve	Maximum Displacement, $\underline{d}$	Natural Frequency, f_n
Linear		$d = \frac{2h}{G_m - 1}$	$f_n = \frac{1}{2\pi} \sqrt{\frac{g(G_m^2 - 1)}{2h}}$
Cubic	 $P = kx + rx^3$	d is slightly $> \frac{2h}{G_m - 1}$ if $r > 0$ and $< \frac{2h}{G_m - 1}$ if $r < 0$	f_n depends on the value of h. If $h >> d$, then f_n is slightly $< \frac{1}{2\pi}\sqrt{\frac{g(G_m^2 - 1)}{2h}}$ if $r > 0$; and f_n is slightly $> \frac{1}{2\pi}\sqrt{\frac{g(G_m^2 - 1)}{2h}}$ if $r < 0$.
Preloaded linear	 $P = P_0 + kx$ for $x > 0$; $P = -P_0 + kx$ for $x < 0$	$d = \frac{2h}{\frac{P_0}{mg} + G_m - 1}$	Let C denote $\sqrt{(G_m mg)^2 - (P_0 - mg)^2}$. Then $f_n = [C/m\sqrt{2gh}] \div \{\ln(C + P_0 - mg) - \ln(C - P_0 + mg) + \ln(C + P_0 + mg) - \ln(C - P_0 - mg)\}$
Preloaded linear with zero slope	 $P = P_0$ for $x > 0$; $P = -P_0$ for $x < 0$	$d = \frac{h}{G_m}$	$f_n = \frac{1}{4}\sqrt{\frac{g}{2h}} \frac{G_m(G_m + 2)}{G_m + 1}$
Tangential	 $P = \frac{2k_0 d_b}{\pi} \tan \frac{\pi x}{2d_b}$	For G_m large, $d = \frac{2.76\, h}{G_m}$. For other values of G_m, see Figure 8	For G_m large, $f_n = 0.0715\, G_m \sqrt{\frac{g}{h}}$. For other values of G_m, see Figure 9

This equation sets the mass times the acceleration equal to the sum of the forces on $\underline{m}$: $-P$, the force by the spring, and $\underline{mg}$, the force due to gravity.

Therefore, $-mG_m g = -P_{max} + mg$ (3)

and $(G_m + 1)\, mg = P_{max}$. (4)

Also, since there is no kinetic energy at $x = d$, loss of potential energy equals energy stored in the spring; or

$$mg\,(h + d) = \int_0^d P\, dx. \tag{5}$$

Linear Spring

$$P = kx$$

where k = spring constant.

Substituting into Equation (5), we obtain

$$mg\,(h + d) = \int_0^d kx\, dx = \frac{kd^2}{2}. \tag{6}$$

From Equation (4), $P_{max} = kd = (G_m + 1)\, mg$.

Substitute this expression for $\underline{kd}$ into Equation (6).

$$mg\,(h + d) = (G_m + 1)\, mg\, \frac{d}{2},$$

$$\therefore\; h = \frac{1}{2}(G_m - 1)\, d. \tag{7}$$

Denote $(G_m - 1)$ by G; then

$$d = \frac{2h}{G},$$

and by Equation (1)

$$e = 2.$$

The natural frequency can be found from the same basic equations.

Solve Equation (6) for d.

$$d = \frac{mg + \sqrt{mg\,(mg + 2kh)}}{k}. \tag{8}$$

Solve Equation (7) for d.

$$d = \frac{2h}{G_m - 1}. \tag{9}$$

Equate these two expressions and solve for m/k.

$$\left(\frac{m}{k}\right) g + \sqrt{\left(\frac{m}{k}\right)^2 g^2 + \left(\frac{m}{k}\right) 2gh} = \frac{2h}{G_m - 1}.$$

Transposing and squaring,

$$\left(\frac{m}{k}\right)^2 g^2 + \left(\frac{m}{k}\right) 2gh = \left[\frac{2h}{G_m - 1} - \left(\frac{m}{k}\right) g\right]^2.$$

Or,

$$\left(\frac{m}{k}\right)\left(2gh + \frac{4gh}{G_m - 1}\right) - \left(\frac{2h}{G_m - 1}\right)^2 = 0,$$

which gives

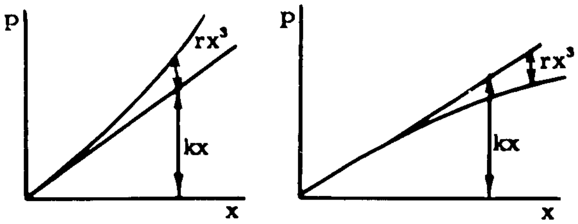
$$\left(\frac{m}{k}\right) = \frac{2h}{g(G_m^2 - 1)}. \qquad (10)$$

Substitute this into the formula for the natural frequency,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}, \qquad (11)$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g(G_m^2 - 1)}{2h}}. \qquad (12)$$

Cubic Spring



r positive   r negative

Figure 3 - Static load displacement curve for cubic spring

$$P = kx + rx^3, \qquad (13)$$

where k and r are constants and r is small.

Substitute P into Equation (5).

$$mg\,(h + d) = \int_0^d (kx + rx^3)\,dx = \frac{kd^2}{2} + \frac{rd^4}{4}. \quad (14)$$

From Equations (4) and (13),

$$P_{max} = kd + rd^3 = (G_m + 1)\,mg, \qquad (15)$$

$$\therefore\ kd = (G_m + 1)\,mg - rd^3.$$

Substitute kd into Equation (14).

$$mg\,(h + d) = \left[(G_m + 1)\,mg - rd^3\right]\frac{d}{2} + \frac{rd^4}{4}.$$

Or

$$h = -d + \frac{d}{2}\left[(G_m + 1) - \frac{rd^3}{mg}\right] + \frac{rd^4}{4mg},$$

$$h = \frac{d}{2}\left[-2 + (G_m + 1) - \frac{rd^3}{mg} + \frac{rd^3}{2mg}\right],$$

$$h = \frac{d}{2}\left[(G_m - 1) - \frac{rd^3}{2mg}\right]. \qquad (16)$$

That is,

$$d = \frac{2h}{(G_m - 1) - \frac{rd^3}{2mg}}. \qquad (17)$$

Thus the value of d is slightly larger or smaller than its value for a linear spring with the same G_m and h, according as r is positive or negative, respectively.

The natural frequency of any spring without friction, for small amplitudes (Reference 2), is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{\left.\frac{\partial P}{\partial x}\right|_{x_e}}{m}}, \qquad (18)$$

where $(\partial P/\partial x)|_{x_e}$ is the slope of the static load displacement curve at the equilibrium position, x_e. (For a linear spring, $(\partial P/\partial x) \equiv k$.)

For a cubic spring, by Equation (13),

$$\left.\frac{\partial P}{\partial x}\right|_{x_e} = k + 3rx_e^2. \qquad (19)$$

Solving Equation (14) for k/m gives

$$\frac{k}{m} = \frac{2g\,(h + d)}{d^2} - \frac{rd^2}{2m}.$$

Substituting into Equation (19) and then into (18) gives

$$f_n = \frac{1}{2\pi}\sqrt{\frac{2g\,(h + d)}{d^2} - \frac{rd^2}{2m} + \frac{3rx_e^2}{m}}. \qquad (20)$$

If r^3 and r to higher powers than 3 are considered negligibly small, then it can be shown that

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m} + 3rm\left(\frac{g}{k}\right)^2\left[1 - \frac{9}{2}r\frac{(mg)^2}{k^3}\right]}.$$

This is not, however, a practical form for comparison; to compare the frequency with that of linear spring with the same h and G_m, Equation (20) will be used. Whether this frequency is larger or smaller than that for a linear spring depends on the relative values of h and d.

Therefore, the frequencies will be compared for the case $h >> d$. To do this it will first be proved that the second and third terms in Equation (20), taken together, have the opposite sign from r.

Solving Equation (14) for $d^2/2$ gives

$$\frac{d^2}{2} = \frac{mg\,(h+d)}{k} - \frac{rd^4}{4k} \approx \frac{mgh}{k}, \tag{21}$$

since d and r are small.

Also, at equilibrium, the spring force balances the weight; thus

$$kx_e + rx_e^3 = mg,$$

or

$$x_e \approx \frac{mg}{k}, \tag{22}$$

since r is small.

After substituting Equations (21) and (22) into (20), the second and third terms of (20) become:

$$-\frac{r}{m}\left(\frac{mgh}{k}\right) + \frac{3r}{m}\left(\frac{mg}{k}\right)x_e$$

$$= \frac{rg}{k}\,(-h + 3x_e),$$

but since h is large, this expression has sign opposite to that of r.

Therefore, from Equation (20),

$$f_n < \text{ or } > \frac{1}{2\pi}\sqrt{\frac{2g\,(h+d)}{d^2}} \tag{23}$$

according as r is positive or negative, respectively.

This d (for a cubic spring) is larger or smaller than d for a linear spring, according as r is positive or negative, by Equation (17). Therefore, according as r is positive or negative, f_n for a cubic spring is smaller or larger than the natural frequency of a linear spring with the same h and G_m; because Equation (23) gives the frequency for a linear spring, since it is Equation (20) with r set equal to zero.

That is (using Equation (12) for the frequency of a linear spring),

$$f_n < \text{ or } > \frac{1}{2\pi}\sqrt{\frac{g\,(G_m^2 - 1)}{2h}},$$

according as r is positive or negative, provided $h >> d$.

Preloaded Linear Spring

A pair of linear springs may be preloaded to give the curve shown in Table 1.

$$P = P_o + kx \qquad \text{for } x \text{ positive,} \tag{24}$$

$$P = -P_o + kx \qquad \text{for } x \text{ negative,} \tag{25}$$

$$\therefore P_{max} = P_o + kd.$$

From Equation (4),

$$(G_m + 1)\,mg = P_o + kd. \tag{26}$$

From Equations (5) and (24),

$$mg\,(h+d) = \int_0^d (P_o + kx)\,dx = P_o d + \frac{kd^2}{2}. \tag{27}$$

Solving for P_o gives

$$P_o = \frac{mg\,(h+d)}{d} - \frac{kd}{2}. \tag{28}$$

Substituting P_o into Equation (26) and solving for G_m gives

$$G_m = \frac{kd}{2mg} + \frac{h}{d}. \tag{29}$$

For a linear spring in which G_m is a minimum for a given m, h, and d, we find from Equation (29) that G_m will be minimum when $k = 0$. Then (29) gives

$$h = G_m d, \tag{30}$$

or

$$d = \frac{h}{G_m}, \tag{31}$$

and thus, by comparison with Equation (1),

$$e = 1.$$

It can be shown that if $k \neq 0$,

$$d = \frac{2h}{\dfrac{P_o}{mg} + (G_m - 1)}.$$

This expression reduces to Equation (31) when $k = 0$ (i.e., $P_o/mg = G_m + 1$, from Equation (26)) is substituted.

It can be shown that when $k \neq 0$, and $\sqrt{(G_m\,mg)^2 - (P_o - mg)^2}$ is denoted by C,

$$f_n = C/\left(m\sqrt{2gh}\right) \div \Big[\ln\,(C + P_o - mg) - \ln\,(C - P_o + mg) + \ln\,(C + P_o + mg) - \ln\,(C - P_o - mg)\Big].$$

As $k \rightarrow 0$ (i.e., $P_o/mg \rightarrow G_m + 1$), it can be shown that

$$f_n \longrightarrow \frac{1}{4}\sqrt{\frac{g}{2h}}\ \frac{G_m(G_m+2)}{G_m+1}, \qquad (32)$$

For simplicity, this will not be done here. Equation (32) only will be derived directly from the assumption that $k = 0$.

After $\underline{m}$ is dropped, $\underline{x}$ is at first positive (Figure 4). Let $t = 0$ at the moment of landing. Then $x|_0 = 0$ and $\dot{x}|_0 = \sqrt{2gh}$.

Substituting Equation (24) (with $k = 0$) into Equation (2) gives

$$m\ddot{x} = -P_0 + mg.$$

Integrating,

$$\dot{x} = \left(-\frac{P_0}{m} + g\right)t + \sqrt{2gh}$$

and

$$x = \left(-\frac{P_0}{m} + g\right)\frac{t^2}{2} + \sqrt{2gh}\,t. \qquad (33)$$

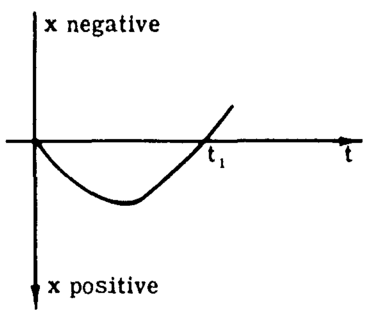


Figure 4

Let t_1 be the time when $\underline{x}$ returns to zero. Substituting $x = 0$ and $t = t_1$ into Equation (33) and solving for t_1, we obtain

$$t_1 = \frac{2\sqrt{2gh}}{\frac{P_0}{m} - g}.$$

When $\underline{x}$ becomes negative (Figure 5), the force P is $(-P_0 + kx)$. Let a new coordinate $\underline{t}$ be zero the instant $\underline{x}$ becomes negative; by conservation of energy, the velocity is then $-\sqrt{2gh}$.

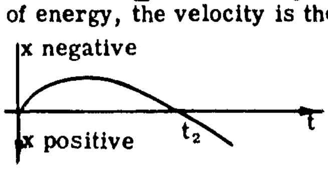


Figure 5

By Equation (2),

$$m\ddot{x} = P_0 + mg,$$

$$\dot{x} = \left(\frac{P_0}{m} + g\right)t - \sqrt{2gh},$$

$$x = \left(\frac{P_0}{m} + g\right)\frac{t^2}{2} - \sqrt{2gh}\,t. \qquad (34)$$

Let t_2 be the time $\underline{x}$ returns to zero; solving Equation (34) for t_2 (when $x = 0$) gives

$$t_2 = \frac{2\sqrt{2gh}}{\frac{P_0}{m} + g}. \qquad (35)$$

The period of the motion is, by definition, $t_1 + t_2$.

$$t_1 + t_2 = 2\sqrt{2gh}\left[\frac{1}{\frac{P_0}{m} - g} + \frac{1}{\frac{P_0}{m} + g}\right]. \qquad (35)$$

But substituting $k = 0$ into Equation (26) gives

$$P_0 = mg\,(G_m + 1).$$

Substituting this into Equation (35) gives

$$\text{Period} = \frac{4\sqrt{2gh}\,(G_m + 1)}{g\,G_m\,(G_m + 2)},$$

$$\therefore f_n = \frac{1}{\text{Period}} = \frac{1}{4}\sqrt{\frac{g}{2h}}\ \frac{G_m(G_m+2)}{G_m+1}.$$

Equation (35) shows that for a given spring, the period of motion depends on $\underline{h}$. The frequency is different from that for a linear spring (Equation (12)), because—although it depends on $\underline{h}$—the frequency of a linear spring is $(1/2\pi)\sqrt{(k/m)}$, regardless of $\underline{h}$.

Thus, the natural frequency of a given preloaded spring depends on $\underline{h}$, or on its initial velocity, $\sqrt{2gh}$. That is, as far as the avoidance of resonant conditions is concerned, the natural frequency of the preloaded spring has no significance.

Tangential Spring

$$P = \frac{2k_0 d_b}{\pi}\tan\frac{\pi x}{2d_b} \qquad (36)$$

where k_0 = constant
d_b = deflection at which the spring bottoms.

From Equation (4),

$$(G_m + 1)\, mg = \frac{2k_o d_b}{\pi} \tan \frac{\pi d}{2d_b}. \qquad (37)$$

Denote $\pi d/2d_b$ by α.

Then

$$G_m = \frac{2k_o d_b}{\pi mg} \tan \alpha - 1. \qquad (38)$$

From Equation (5),

$$mg\,(h + d) = \int_0^d \frac{2k_o d_b}{\pi} \tan \frac{\pi x}{2d_b}\, dx, \qquad (39)$$

$$mg\,(h + d) = k_o \left(\frac{2d_b}{\pi}\right)^2 \ln \sec \alpha.$$

From this equation, one can define

$$f(k_o, d) \equiv 0 \equiv k_o \left(\frac{2d_b}{\pi}\right)^2 \ln \sec \alpha - mg(h+d). \quad (40)$$

Only one type of tangential spring will be considered; namely, the one with minimum G_m, given $\underline{m}$, d_b, and $\underline{h}$. To find the value of k_o for this spring, notice that Equation (38) gives G_m as a function of k_o and $\underline{d}$; and (40) gives $\underline{d}$ as a function of k_o. Therefore,

$$\frac{dG_m}{dk_o} = \left.\frac{\partial G_m}{\partial k_o}\right|_d + \left.\frac{\partial G_m}{\partial d}\right|_{k_o} \frac{dd}{dk_o} = 0, \qquad (41)$$

where $(\partial G_m/\partial k_o)|_d$ denotes the partial derivative with respect to k_o, keeping $\underline{d}$ constant (as well as $\underline{m}$, d_b, and $\underline{h}$).

By a well-known theorem, Equation (40) gives

$$\frac{dd}{dk_o} = -\frac{\dfrac{\partial f(k_o,d)}{\partial k_o}}{\dfrac{\partial f(k_o,d)}{\partial d}}$$

$$\frac{dd}{dk_o} = -\frac{\left(\dfrac{2d_b}{\pi}\right)^2 \ln \sec \alpha}{k_o\left(\dfrac{2d_b}{\pi}\right) \tan \alpha - mg}.$$

Substituting this into Equation (41) and using (38) gives

$$\frac{dG}{dk_o} = \frac{2d_b}{\pi mg} \tan \alpha + \left(\frac{k_o}{mg} \sec^2 \alpha\right)\left[-\frac{\left(\dfrac{2d_b}{\pi}\right)^2 \ln \sec \alpha}{k_o\left(\dfrac{2d_b}{\pi}\right)\tan\alpha - mg}\right] = 0.$$

Therefore,

$$2 \tan \alpha = \frac{2k_o d_b}{mg\pi} \; \frac{\sec^2\alpha \ln \sec^2\alpha}{\dfrac{2k_o d_b}{mg\pi} \tan \alpha - 1}. \qquad (42)$$

By Equation (38),

$$\frac{2k_o d_b}{mg\pi} = \frac{G_m + 1}{\tan \alpha}. \qquad (43)$$

Substituting into Equation (42) gives

$$2 \tan \alpha = \frac{G_m + 1}{\tan \alpha} \; \frac{\sec^2\alpha \ln \sec^2\alpha}{(G_m + 1) - 1}.$$

Denote $\tan^2\alpha$ by β and $\sec^2\alpha$ by $(1 + \beta)$. Then

$$2\beta = \frac{G_m + 1}{G_m} (1 + \beta) \ln (1 + \beta). \qquad (44)$$

If G_m is given, this equation can be solved for β. Figure 6 gives some corresponding values of G_m and β.

G_m	β
1	0
1.1	.11
2	.82
5	2.1
7	2.5
10	2.9
40	3.6
1000	3.91
∞	3.95

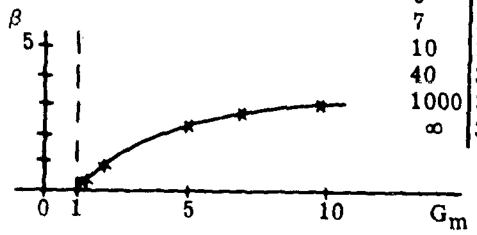


Figure 6 - (Equation 44)

By definition of α,

$$d = \frac{2d_b \alpha}{\pi}. \qquad (45)$$

From Equation (37),

$$\frac{2k_o d_b}{\pi} = \frac{(G_m + 1)\ mg}{\tan\alpha}. \tag{46}$$

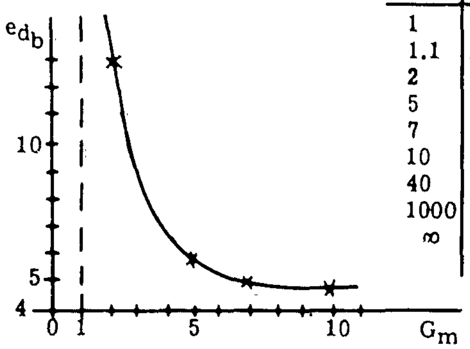
Substituting Equations (45) and (46) into (39) and solving for d_b gives

$$d_b = \frac{\frac{\pi}{2}\ h}{\frac{(G_m + 1)}{\tan\alpha}\ln\sec\alpha - \alpha}.$$

Let

$$d_b = \frac{e_{d_b}\ h}{G_m}, \tag{47}$$

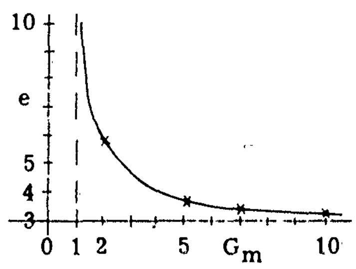
define e_{d_b}. Values of e_{d_b} are shown in Figure 7.



G_m	e_{d_b}
1	∞
1.1	172
2	12.6
5	5.75
7	5.09
10	4.67
40	4.10
1000	3.92
∞	3.91

Figure 7

Now d can be found from d_b above and Equation (45). Then $\underline{e}$, plotted in Figure 8, can be found from Equation (1).



G_m	e
1	∞
1.1	35.2
2	5.94
5	3.55
7	3.27
10	3.09
40	2.84
1000	2.75
∞	2.75

Figure 8 - The elasticity parameter of a tangential spring, which for a given $\underline{m}$, d_b, and $\underline{h}$ values has a minimum G_m value

The natural frequency will now be found. Note that at equilibrium, the spring force balances the weight. Using Equation (36)

$$\frac{2k_o d_b}{\pi}\tan\frac{\pi x_e}{2d_b} = mg. \tag{48}$$

By Equation (36),

$$\left.\frac{\partial P}{\partial x}\right|_{x_e} = k_o\left(1 + \tan^2\frac{\pi x_e}{2d_b}\right). \tag{49}$$

Substituting Equation (48) into (49) gives

$$\left.\frac{\partial P}{\partial x}\right|_{x_e} = k_o\left[1 + \left(\frac{mg\pi}{2k_o d_b}\right)^2\right].$$

Then Equation (18) becomes

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_o}{m}\left[1 + \left(\frac{mg\pi}{2k_o d_b}\right)^2\right]}. \tag{50}$$

Write Equation (43) as

$$k_o d_b = \frac{\pi mg\ (G_m + 1)}{2\tan\alpha}. \tag{51}$$

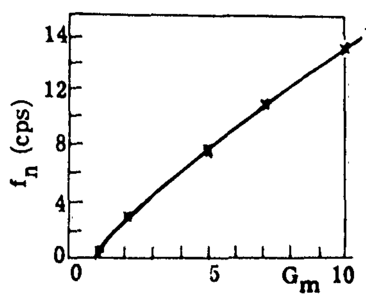
Substituting Equation (47) into (51),

$$k_o = \frac{\pi mg\ (G_m + 1)\ G_m}{2\tan\alpha\ e_{d_b}\ h}. \tag{52}$$

Substituting Equations (51) and (52) into (50) gives

$$f_n = \frac{1}{2\pi}\sqrt{\frac{\pi g\ (G_m + 1)\ G_m}{2\tan\alpha\ e_{d_b}\ h}\left[1 + \left(\frac{\tan\alpha}{G_m + 1}\right)^2\right]}. \tag{53}$$

Corresponding values of G_m, $\tan\alpha (=\sqrt{\beta})$, and e_{d_b} are taken from Figures 6 and 7 and substituted into Equation (53). The resulting values of f_n are shown in Figure 9.



G_m	f_n
1	$0/\sqrt{h}$
1.1	$0.793/\sqrt{h}$
2	$2.96/\sqrt{h}$
5	$7.61/\sqrt{h}$
7	$10.5/\sqrt{h}$
10	$14.7/\sqrt{h}$
40	$56.5/\sqrt{h}$
1000	$1400/\sqrt{h}$
∞	∞

Figure 9

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EVALUATING SHOCK AND VIBRATION RESISTANCE IN TERMS OF LABORATORY TEST FAILURES

K. E. Woodward, NRL

Two hundred seventy equipments were subjected to laboratory vibration and shock tests. Resulting damage to different components is presented in tabular form. Equipment failures indicated by various shock machines are shown graphically. The effectiveness of shock mounts is discussed.

The ultimate goal in the design and mounting of an equipment is to produce a system which will function satisfactorily regardless of the vibration or shock conditions present in the final installation, excepting, of course, some extraordinary situation such as a direct hit in warfare. There are two possible extremes in the approach to obtaining such performance. One extreme is to mount the unit on a low-frequency system capable of large deflections which will isolate the equipment from vibration as well as shock forces. The other is to build the equipment rugged enough so that fatigue limits, ultimate strengths, critical displacements, or other limiting criteria are not exceeded, irrespective of the type of mounting.

The first approach is impracticable, particularly for Navy application, because of the large clearances required. With clearances limited, low-frequency systems are not desirable for shipboard use because of collision or anticipated "hard-bottoming" effects resulting from shock forces. Therefore, the Navy follows an intermediate course in which relatively stiff mounts are used. These mounts aggravate the normal vibrations encountered below approximately 25 cps but afford some protection for the high-frequency components of shock.

Over a period of years the Navy has devised specifications and testing machines which attempt to duplicate in the laboratory the maximum shock and vibration conditions expected on shipboard for which protection is required. If, then, one considers the testing specifications as the criterion of design, it would be well to examine the failures and successes of previously tested equipments to determine factors of good and bad design.

For the past nine or ten years the Naval Research Laboratory has been performing shock and vibration approval tests on equipments submitted by various manufacturers under contract to the Navy. Recently, an investigation was conducted on 270 individual equipments tested between November 1944 and August 1952. A summary of the vibration and shock specifications governing the tests is shown in Tables 1 and 2. The equipments investigated ranged in weight from 3025 lb to less than 1 oz, and in size from a radar antenna measuring 17 ft across the tips of the reflector to small 30-ampere ferrule-type fuse clips. Of the 270 units considered, 107 contained electron tubes. The investigation covered approximately 40 percent of all such tests performed by the Shock and Vibration Group of the Laboratory. The equipments themselves were not available for the study; but test reports describing the equipments and the damages were, and these were used as source material for the investigation.

A brief explanation of the manner in which the study was conducted is in order so that use of term "damage" may be clarified. The term is applied to any change of conditions in the

TABLE 1
Summary of Specifications

Spec.	Date	Vibration Characteristics		Vibration Tests						Light-Weight Shock Tests	
				Initial Vibration Test			Final Vibration Test			Height of Hammer Drop (Ft.)	
		Freq. R.* (cps)	Exc. (mils)	Freq. R.* (cps)	Exc. (mils)	Duration	Freq. R. (cps)	Exc. (mils)	Duration	Both Hor. Directions	Vertical Direction
MIL-T-945A	Mar 50			10 to 33 Cycling 10-33-10	Min: 54 Max: 66	15 min ½ min cycle	10 to 55 Cycling 10-55-10	Min: 24 Max: 36	15 min 1 min / cycle	1-2-3	2-3-4-
MIL-S-901	Nov 49									1-3-5	1-3-5
40T9	Dec 48	5 to 33	Min: 6 Max: 16	5 to 23	Min: 48 Max: 72	3 min. each frequency,	Worst frequency 5 to 23	Min: 48 Max: 72	120 min	1-2-3	1-2-3
66S3	Sept 45									1-3-5	1-3-5
16T35(RE)	July 45			5 to 23	Min: 48 Max: 72	3 min each frequency,	23 to 55	Min: 6 Max: 16	1 min each frequency	1-2-3	2-3-4
RE 9284B	Aug 44	5 to 33	Min: 6 Max: 16	5 to 23	Min: 48 Max: 72	3 min each frequency	Worst frequency 5 to 23	Min: 48 Max: 72	120 min	1-2-3	2-3-4
17E13(INT)	June 44									1-3-5	1-3-5
RE13A825C	May 44			10 to 55 Cycling 10-55-10,	Min: 48 Max: 72	30 min 1-3 min cycle	15 min each res't frequency	Min: 48 Max: 72			

* Freq. Range

Note: Additional test for MIL-T-945A After the completion of the above tests, the equipment shall then be vibrated for a period of 3 minutes at each of the four most severe resonant frequencies, at excursions corresponding to each frequency range, and in the plane observed to give the most severe reaction

TABLE 2
Medium-Weight Equipment Shock Tests

Table Weight (lb)	Height of Hammer Drop (ft)					
	Test A			Test B		
	Group I (3-in. TT*)	Group II (3-in. TT)	Group III (1½-in. TT)	Group I (3-in. TT)	Group II (3-in. TT)	Group III (1½-in. TT)
250 - 999	0.75	1.75	1.75	0.75	1.00	1.00
1000 - 1999	1.00	2.00	2.00	0.75	1.25	1.25
2000 - 2999	1.25	2.25	2.25	0.75	1.50	1.50
3000 - 3499	1.50	2.50	2.50	1.00	1.75	1.75
3500 - 3999	1.75	2.75	2.75	1.00	2.00	2.00
4000 - 4199	2.00	3.00	3.00	1.25	2.25	2.25
4200 - 4399	2.00	3.25	3.25	1.25	2.25	2.25
4400 - 4599	2.00	3.50	3.50	1.25	2.25	2.25
4600 - 4799	2.25	3.75	3.75	1.50	2.50	2.50
4800 - 4999	2.25	4.00	4.00	1.75	2.75	2.75
5000 - 5199	2.50	4.50	4.50	1.75	2.75	2.75
5200 - 5399	2.50	5.00	5.00	1.75	2.75	2.75
5400 - 5600	2.50	5.50	5.50	1.75	2.75	2.75
	Test C			Test D		
	Group I (3-in. TT)	Group II (1½-in. TT)	Group III (3-in. TT)	Group I (3-in. TT)	Group II (1½-in. TT)	Group III (3-in. TT)
400 - 499	0.85	1.85	1.85	1.25	2.75	2.75
500 - 1099	1.00	1.85	1.85	1.50	2.75	2.75
1100 - 1499	1.00	2.00	2.00	1.50	3.00	3.00
1500 - 1699	1.15	2.00	2.00	1.75	3.00	3.00
1700 - 2299	1.15	2.15	2.15	1.75	3.25	3.25
2300 - 2499	1.15	2.35	2.35	1.75	3.50	3.50
2500 - 2799	1.35	2.35	2.35	2.00	3.50	3.50
2800 - 3199	1.35	2.50	2.50	2.00	3.75	3.75
3200 - 3299	1.35	2.65	2.65	2.00	4.00	4.00
3300 - 3499	1.50	2.65	2.65	2.25	4.00	4.00
3500 - 3799	1.50	2.85	2.85	2.25	4.25	4.25
3800 - 3999	1.65	3.00	3.00	2.50	4.50	4.50
4000 - 4199				2.50	4.75	4.75
4000 - 4299	1.85	3.35	3.35			
4200 - 4299				2.75	5.00	5.00
4300 - 4399	1.85	3.50	3.50	2.75	5.25	5.25
4400 - 4500	1.85	3.65	3.65	2.75	5.50	5.50

*Indicates Table Travel

Note: Each test consists of three groups of two blows each with anvil table travel as indicated.
Bureau of Ships Specification 40T9 - Tests A and B
Navy Department Specification 66S3 - Test A
Military Specification MIL-S-901 - Test A
Bureau of Ships Specification RE 9284B - Test C
Bureau of Ships Ad Interim Specification 17E13(INT) - Test D

equipments resulting from the tests or to any deficiency of design which made the units unable to meet specification requirements.

Damages were recorded for primary failures only. If, for example, the mounting bolts on a component failed during a shock blow and the uncaptured component caused secondary damage, the primary damage was recorded as a bolt failure and the secondary damage was not tabulated. The possibility that the secondary damage could have been caused by the shock alone was not eliminated, but the more obvious cause was collision.

In recording damages on such items as fractured fasteners (i.e., screws, rivets) where more than one fastener was employed for a particular function, the damage was recorded as a single fractured fastener. This procedure was necessary because of the lack of clarity in many of the test reports as to the exact number of fasteners involved. Also, in those equipments where replacement components or parts suffered the same damage during subsequent testing as did the original parts, the damage was recorded for the original components only. In this way, undue weight was not given to a damage which resulted from a poor initial design for a particular unit.

The damages suffered by the equipments studied are summarized in Table 3, which lists the number of damages resulting from shock and from vibration, according to components, with the estimated quantity of each component in the equipments tested. Space does not permit an elaboration of the table to show exactly what has failed, but later this year a report showing a complete breakdown of each category of the table will be published by the Naval Research Laboratory (NRL Report 4179). Components not appearing in the table were undamaged by the tests or suffered only a single damage, and were included under miscellaneous categories. In some instances, because of the lack of sufficient information, it was not possible to estimate the quantity of an individual component. The percentages which are shown should be considered as quantities representing the relative "goodness" of components rather than a figure expressing the number of damaged components per 100 tested, since it is possible that more than one damage was tabulated for the same component.

Table 4 is a listing of vacuum-tube failures, showing the estimated number of tubes tested and the number of damages to specific tube types occurring under both shock and vibration. Because of the lack of uniformity in testing conditions, no conclusions were drawn from the table regarding tubes which were good or bad for dynamic situations.

Without becoming involved in the details of specific damages, a number of general observations are made. Table 3 shows a total of 858 damages recorded for 222 shock tests and 844 damages for 229 vibration tests, or a nearly equal number of damages for almost the same number of each type of test. This comparison is made, of course, without regard to type of mounting, type of equipment, or severity of testing conditions. In current specifications, vibration testing normally is more damaging on the average equipment than is shock testing because of revisions to or different interpretations of earlier specifications. Many of the vibration tests conducted from 1944 to 1947, which were included in this study, did little more than quality check components.

The totals are significant for the following reason. Most design engineers, during their first visit to the Laboratory, leave the impression that the majority of them understand shock to be far more damaging than vibration. But for the average equipment tested under current shipboard specifications, this is not true. Unfortunately, in the minds of the uninitiated, the term "shock resistance" minimizes the damaging effects of vibration.

Although the inertial forces acting on the equipments during vibration were low relative to those during shock (normally between 2 and 5 gravity units as compared to 300), the repeated nature of the forces was as damaging as larger shock forces acting for a very short period of time. Vibration not only induced material failures due to fatigue, but the amplification of vibration due to the resonance of the structures and of the shock mounts (if used) caused many components mounted on the structures to malfunction without actually failing.

An equipment well designed for vibration was usually very good for shock. But an equipment that passed shock tests satisfactorily may or may not have been capable of passing vibration tests, because many were too flexible and thus were resonant in the testing frequencies. The flexibility was usually beneficial for shock if collision did not occur.

Perhaps the most deficient areas of design revealed by the study were the chassis, cabinets, and frame structures. Again the difficulty

TABLE 3
Damages to Specific Parts or Components
(All Equipments)

Units	Estimated No. of Units	SHOCK No. of Damages	SHOCK %*	VIBRATION No. of Damages	VIBRATION %*
Ball bearings	520	3	0.6	0	0.0
Cabinet and frame	220	81	36.8	63	28.6
Castings (miscellaneous materials and shapes)	220	4	1.8	1	0.5
Chassis	270	57	21.2	70	26.0
Condensers	10,865	4	0.1	13	0.1
Connectors (cable)	1,300	6	0.5	6	0.5
Control knobs	580	2	0.4	2	0.4
CRO tubes	31	11	35.5	2	6.5
Crystals	110	0	0.0	2	1.8
Fasteners (all types)	104,200	204**	0.2	157**	0.2
Flexible couplings	-	0	-	2	-
Fluorescent lamps	28	5	17.9	3	10.7
Fluorescent lamp sockets	56	4	7.1	0	0.0
Fluorescent lamp starters	28	0	0.0	6	21.4
Gears (miscellaneous types and materials)	590	3	0.5	0	0.0
Hinges	85	1	1.2	5	5.9
Incandescent lamps (50-3000 watts)	24	18	75.0	6	25.0
Incandescent lamp sockets	24	10	41.7	3	12.5
Indicator lamps	430	9	2.1	7	1.6
Latches	90	9	10.0	1	1.1
Locks for front panel controls	290	9	3.1	18	6.2
Meters and indicators	190	23	12.1	18	9.5
Mounting brackets (cantilever type)	70	16	22.9	19	27.2
Mounting yokes (Y type)	7	1	14.3	4	57.2
Panel-mounted fuse holders	210	4	1.9	0	0.0
Pedestals	8	5	62.5	2	25.0
Relays	300	28	9.3	17	5.7
Resistors	13,400	0	0.0	7	0.1
Shafts (control and power transmission)	-	6	-	5	-
Shock mounts	350	12	3.4	5	1.4
Springs (coil and leaf)	-	5	-	10	-
Switches (all types)	800	11	1.4	8	1.0
Terminal strips and component mounting boards	630	3	0.5	1	0.2
Transformers and inductors	930	4	0.4	22	2.4
Tubular structures (welded)	-	11	-	11	-
Vacuum tubes Estimated: "	Shock-2,440 Vib.-2,280	58	2.4	130	5.7
Vacuum tube clamps " "	Shock-2,431 Vib.-2,272	17	0.7	10	0.4
Vacuum tube plate caps " "	Shock- 175 Vib.- 181	8	4.6	4	2.2
Vacuum tube sockets " "	Shock-2,440 Vib.-2,280	0	0.0	3	0.1
Wave-guide sections	60	7	11.7	1	1.7
Wiring	-	22	-	148	-
Miscellaneous Damages					
Bonding failures	-	4	-	0	-
Brittle material failures	-	20	-	2	-
Collision of components	-	36	-	9	-
Parts dependent on frictional clamps or pressfits	-	82	-	22	-
Miscellaneous damages	-	22	-	14	-
Weld failures (miscellaneous)	-	13	-	1	-
Vibration isolated components	4	0	0.0	4	100.0
		858		844	

* Percentages are based on the total estimated components of all equipments investigated. Total shock tests - 222; total vibration tests - 229.

**This figure indicates the total number of instances involving damage to fasteners. It does not represent the total number of fasteners damaged.

TABLE 4
Vacuum Tubes Subjected to Shock and Vibration Tests in Equipments

Tube	SH*	DA*	VI*	DA
OA2	32		39	
OA2/VR150	2		2	
OA3	3		3	
OA3/VR75	2		2	
OB2	19		25	1
OC3	4		4	1
OC3/VR105	6		6	
OC3W	1		1	
OD3	15		14	
OD3/VR150	11		13	
OD3/VR150W	1		1	
OD3W	2		2	
1B3	1		1	
1B27	2			
1B35	1		1	
1B63A	5		5	
1N21B	3		1	
1N22	4			
1N23B	12		12	
1N44	8		8	
1N45	8		8	
1N47	8		8	
1T4	2		2	
1V2	2		2	
1Z2	9	1	8	
2A3	2		2	
2AP1	2		1	
2BP1	6		6	1
2C36	1		1	
2C39A			2	2
2C40	1		1	
2C51	53		63	2
2C53	1		1	
2D21	29		31	1
2D21W	1		1	
2J41			1	
2J51	1		1	
2K25	4	1	3	2
2K28			3	
2K29	1			
2K45	1		2	1
2X2	1		1	
2X2A	6	1	5	
3A4	4		4	1
3A5	2		5	
3B24	13		13	1
3B24W	8		10	
3B26	7		7	1
3B28	22	3	22	5
3BP1	1		1	
3C23	2		4	1
3E29	1		1	
3JP1	1		1	
3KP1	1		1	
3Q4			1	
4-65A	1		1	
4-400A	3	3	3	
4B31	2		2	
4B32	2		2	2
4D21	4		4	
4J36-41	1			

Tube	SH	DA	VI	DA
4X150A	3		5	2
5C22	4	1	3	
5CP1A	1	1		
5CP7	1	1	1	
5CP12	5		5	1
5FP7			1	
5R4GY	73		69	9
5R4WGY	13	2	16	
5RP2	1		1	1
5SP1	2		2	
5U4G	12	1	12	4
5U4WG	2		2	
5V4			1	1
5V4G	2		2	
5Y3	1		1	
5Y3GT	4		6	
5Y3GT/G	3		3	
5Z3	1		1	
5Z4	1		1	
6-8B	1		1	
6-11			1	
6-13	1	1		
6AB7	4		5	
6AC7	7		7	1
6AC7W	5		1	
6AG5	11		7	
6AG7	47	3	31	2
6AH6	38	1	35	1
6AH	1		1	1
6AK5	89	2	80	4
6AK5W	10		10	1
6AK6	20		20	
6AL5	76	3	72	2
6AL5W	18		23	
6AN5	20	2	18	
6AQ5	36		34	5
6AQ6	3		3	
6AR6	4		4	2
6AS6	38		35	2
6AS7	9		7	6
6AS7G	6		8	
6AT6	1		1	
6AU6	21		23	
6B4G	7		8	
6BA6	16		16	2
6BE6	8		8	1
6BM6	1		1	
6C4	18		26	
6C5	2		1	
6CB6	1		1	
6D4	5		5	
6D6	2		2	
6E5	1		1	
6F4	3		3	
6F6	2		1	
6H6	21		13	2
6H6WGT	3		3	
6J5	17	5	5	
6J5GT	1			

Tube	SH	DA	VI	DA
6J6	82		86	3
6J7			8	
6K6GT	1		1	
6K7	5		6	
6L6	21		7	
6L6GA	8	1	8	1
6L6GAW	3		3	
6L6W	4		4	4
6L7	1			
6NO30	1	1	1	
6N7	7		3	
6N7GT	1		1	
6R7	1			
6SA7	7		7	
6SG7	8		4	
6SH7	8		9	
6SJ7	35		35	1
6SJ7GT	1		1	1
6SK7	7		8	1
6SK7W	2		2	
6SL7	5	1	3	
6SL7GT	10		13	
6SL7W	20		17	
6SL6WGT	2		2	
6SN7	25		23	
6SN7GT	8		15	
6SN7W	43	3	34	
6SN7WGT	4		4	
6SQ7			1	
6SQ7GT/G	1		1	
6SU7	2		2	
6V6	5	1	8	1
6V6GT	21		20	
6V6GT/G	4		6	
6X4	25		25	6
6X4W	3		5	
6X5	2		2	1
6X5GT	5		9	
6X5GT/G	6	1	5	
6Y6	1		1	1
6Y6G	66		70	
7F8	10		6	
7F8W	20			
10KP7	1			
12A6	6		4	
12AT7	54		67	
12AU7	189		142	5
12AX7	35	1	33	4
12BA6	2		2	
12DP7	1		1	
12H6	7	1	7	
12J5WGT	1		1	
12J7	4		4	
12SA7	1		3	
12SG7	8		6	
12SJ7	8		7	
12SK7	3		3	
12SL7	2		2	
12SL7GT	5		9	
12SN7	2		2	
12SN7GT	4		4	

Tube	SH	DA	VI	DA
12SQ7	2			
25L6GT	2		2	
25X6GT/G	1		1	
25Z6GT	1		1	
25Z6GT/G	1		1	
28D7	3	1	3	
36	1			
38	1			
39/44	3			
84	1		1	
100TH	3		3	3
VR105-30	5		6	
323A	2		2	
393A	4		4	
420A	1		1	
705A	4			
715B	2		2	
719A	1		1	
803	2		2	
807	9	2	10	5
807W	1		1	1
810	2	1	2	2
811			4	
813			2	
829B	5			
832	1			
832A	1	1		
837			2	
860	4		4	
884	3		3	
913	1		1	
927	1		1	
954			1	
955			1	
956			1	
991	5		4	
K1052P2	1		1	
1614	2			
1625			4	
1629	1		1	
2050	4		7	2
2051			1	1
5517	1		1	
5651	9		8	
5654	58		70	
5670	7		7	
5686	6		6	
5687	51	1	52	3
5693	6		6	
5696	6			
5721	3		3	
5725	26		26	
5726	15		15	
5727	6		6	
5751	9		9	
5755	4		4	
5763	1		1	
5780	2		2	
5814	32		32	
5879	1		1	
5932	14		14	
5933	10		10	
9003	9	1	9	
Unknown	208	9	126	12

*SH - Shock tested
DA - Damaged
VI - Vibration tested

Total: Shock tested - 2440
Damaged - 58
Vibration tested - 2280
Damaged - 130

normally resulted from a lack of sufficient structural stiffness, which caused low resonant points and high transmissibility ratios. Poor structural design not only caused damage to the structure but reflected in poor component performance under both vibration and shock.

Relative to structural design, a very high percentage of the larger shock-mounted, box-shaped equipments having base and bulkhead mounting systems and a depth-to-breadth ratio greater than one, had natural frequencies either in, or only slightly above, testing ranges. The same was true for equipments shock mounted on the base only which had heights greater than the minimum distance between shock mounts. In unshocked-mounted equipments these ratios were exceeded somewhat before difficulty of this nature arose.

Designwise, every effort possible should have been made to locate the heavier items, such as transformers and chokes, as near to the bulkheads and decks as possible. The more fragile components (vacuum tubes, etc.) should have been grouped in the center of the chassis, and the other more rugged items (resistors and condensers or small transformers) should have been located around the edges of the chassis. By this means, the vacuum tubes would have benefited from the greater deflections under shock, and the heavier, more rugged components from the stiffer mountings for vibration.

The ultimate in design of packaged equipments would be to locate all transformers on the bottom chassis and the vacuum tubes and other components on the upper chassis. In this way, the flexibility of the chassis could then be varied so that shock protection would be offered to the components by virtue of the chassis design. Unnecessary stiffness is detrimental to high damage resistance, particularly for electronic equipments.

Viewing the over-all damage picture of future equipments, approximately 90 percent of all damages resulting either from vibration or shock can be eliminated from equipments of current design if the components are chosen, modified (if necessary), and mounted with consideration for the damages suffered by previously tested equipments. The remaining 10 percent of the damages can be decreased only by a complete series of ruggedized vacuum tubes and shockproof relays. In the test, vacuum tubes suffered approximately twice the number of failures under vibration as under shock, whereas shock produced more relay failures than vibration. Percentagewise, vacuum tube failures were low relative to other component failures, but the total number of tube failures was high. A possible solution for the malperformance of relays appears to be the use of rotary types in which all forces and moments, with the exception of rotational moments, can be balanced.

As shown in the summarized damage table, some components suffered more from shock and others more from vibration. Still others were damaged about equally under both.

Tables 5 and 6 show the test results i.e., whether the units were satisfactory or unsatisfactory, on equipments tested on shock mounts and those tested solid mounted (i.e., on metal feet). Table 5 considers all of the equipment investigated, whereas Table 6 shows only the results for units containing vacuum tubes. The satisfactory or unsatisfactory conclusions were arranged according to testing specifications and were the conclusions recorded in the test reports. Here it must be pointed out that equal numbers of equipments were not studied under each of the specifications, but sufficient numbers of equipments were investigated for definite trends to be observable.

An important conclusion relative to test methods is drawn from these tabulations. Prior to 1947 the tests were generally conducted on a "go, no-go" basis. Equipments received from the manufacturer were subjected to tests under the applicable specifications without extensive design changes, even though weaknesses were revealed by preliminary tests. For example, if an equipment was improperly shock mounted, an effort usually was made to adjust the stiffness of the mounts in order to place the lowest resonant frequency of the unit above testing frequencies, if possible. Minor adjustments and repairs were made during the course of the tests, but generally nothing more extensive than this was done.

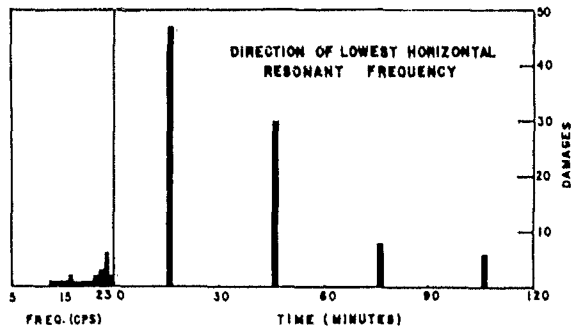
Now, considering the publication dates on the testing specifications, one will observe that for specifications dated later than December 1946 a sharp increase in satisfactory equipments occurs. The increase was due primarily to a change in testing procedures, although some improvement was observed in those equipments submitted by manufacturers who had previously sent units for tests. During 1947 and afterwards, developmental testing came into prominence. In developmental testing, failures or damages resulting from tests are repaired and

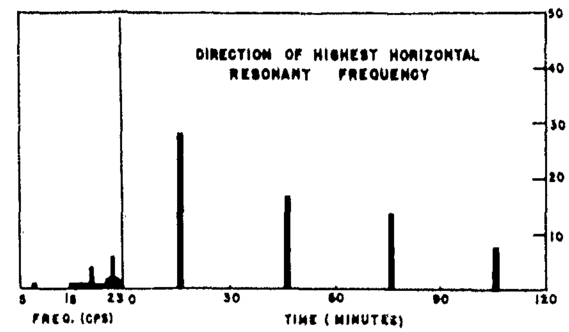
TABLE 5
Results of Tests All Equipments

Specification	Vibration			Shock		
	Percent Satis.	Percent Unsatis.	Number of Tests	Percent Satis.	Percent Unsatis.	Number of Tests
Shock-Mounted Equipments						
Mil-S901	-	-	0	87.0	13.0	23
40T9	78.2	21.8	64	82.1	17.9	39
66S3	-	-	0	84.6	15.4	13
16T35(RE)	100.0	0.0	2	-	-	0
RE 9284B	38.4	61.6	13	53.9	46.1	13
RE 13A825C	50.0	50.0	4	-	-	0
Misc. Spec. & Devised Cond.	50.0	50.0	2	0.0	100.0	1
			85			89
Unshock-Mounted Equipments						
MIL-T-945A	75.0	25.0	4	80.0	20.0	5
MIL-S-901	-	-	0	60.0	40.0	5
40T9	85.4	14.6	82	86.0	14.0	57
66S3	-	-	0	58.3	41.7	12
16T35(RE)	90.0	10.0	10	55.6	44.4	9
RE 9284B	63.2	36.8	19	55.6	44.4	18
17 E 13(INT)	-	-	0	56.0	44.0	25
RE 13A825C	71.4	28.6	7	-	-	0
Misc. Spec. & Devised Cond.	54.5	45.5	22	50.0	50.0	2
			144			133

TABLE 6
Results of Tests Electronic Equipments

Specification	Vibration			Shock		
	Percent Satis.	Percent Unsatis.	Number of Tests	Percent Satis.	Percent Unsatis.	Number of Tests
Shock-Mounted Equipments						
MIL-S-901	-	-	0	100.0	0.0	1
40T9	63.3	36.7	30	80.7	19.3	31
66S3	-	-	0	80.0	20.0	5
RE 9284B	20.0	80.0	10	44.5	55.5	9
RE 13A825C	33.4	66.6	3	-	-	0
Misc. Spec. & Devised Cond.	50.0	50.0	2	0.0	100.0	1
			45			47
Unshock-Mounted Equipments						
MIL-T-945A	75.0	25.0	4	80.0	20.0	5
40T9	87.5	12.5	24	95.0	5.0	20
66S3			0	50.0	50.0	2
16T35(RE)	85.7	14.3	7	50.0	50.0	8
RE 9284B	66.6	33.4	3	20.0	80.0	5
RE 13A825C	50.0	50.0	2	-	-	0
Misc. Spec. & Devised Cond.	55.6	44.4	9	50.0	50.0	2
			49			42





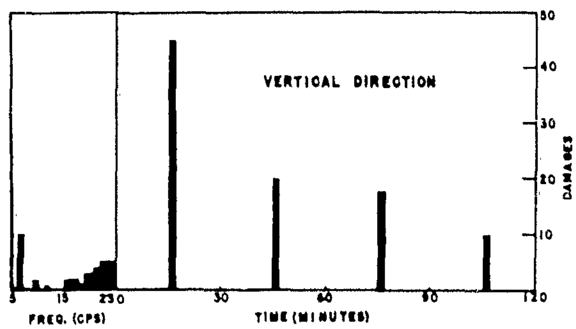


Figure 1 - Vibration damages to equipments tested according to BuShips Specification 40T9

modifications made immediately so that subsequent testing will not usually produce the same damage.

As an illustration, Figure 1 shows damages which occurred to 120 equipments which had satisfactorily passed the vibration tests of BuShips Specification 40T9. This specification was in effect through August 1952 and was the shipboard specification under which most of the developmental tests were conducted. Included in the results are medium- and light-weight electronic and nonelectronic equipments, 50 of which were shock mounted and 70 unshock mounted. The graph does not record all vibration damages, but only those recorded in test reports for which time of occurrence was given. The damages were grouped in half-hour periods, and the totals were placed at the center of each half-hour period of the two-hour constant-frequency test.

It is seen that damages increase with frequency for the variable-frequency test, and that the first half hour of the constant-frequency test produces more damages than any of the following half-hour periods. If these tests had been performed on a "go, no-go" basis, a much larger percentage of units would have proved to be unsatisfactory. By modifying equipments as damages occur, the percentage of satisfactory units is greatly increased, resulting in time and monetary savings to both government and manufacturer and in increased reliability and longer life to the equipment in service.

Another very interesting detail is revealed by expanding the results of the 40T9 tests, as shown in Table 7, under which approximately 50 percent of all of the equipments were tested. Slightly higher satisfactory percentages resulted for light-weight equipments tested without shock mounts, than for those subjected to the tests on shock mounts. This condition was true regardless of the general type of equipment under test, that is, whether or not the equipment was electronic or nonelectronic.

The types of medium-weight equipments tested with and without mounts were generally of a dissimilar nature in construction. The majority of the unshock-mounted units were nonelectronic antenna assemblies with base-mounting systems only, whereas the shock-mounted equipments were normally box-shaped, packaged, electronic units with base and bulkhead mounting arrangements. Low satisfactory percentages resulted for medium-weight, unshock-mounted assemblies, chiefly because of fractures in pedestals. No similar comparison was drawn from these values, as was done for the type of mounting for light-weight equipments, because of this dissimilar nature of the units.

TABLE 7
Equipments Tested According to BuShips Specification 40T9

Equipment	Vibration Tests				Shock Tests			
	Percent Satis.	Percent Unsatis.	No. of Tests	Percent Tests on Elec. Equip.	Percent Satis.	Percent Unsatis.	No. of Tests	Percent Tests on Elec. Equip.
All Equipments								
Shock Mounted								
Light-Weight	85.4	14.6	48	35.4	80.0	20.0	20	70.0
Medium-Weight	56.3	43.7	16	81.3	84.2	15.8	19	89.5
Unshock Mounted								
Light-Weight	88.9	11.1	72	30.6	90.0	10.0	50	36.0
Medium-Weight	60.0	40.0	10	20.0	57.2	42.8	7	28.6
Electronic Equipments Only								
Shock Mounted								
Light-Weight	76.5	23.5	17		78.6	21.4	14	
Medium-Weight	46.1	53.9	13		82.4	17.6	17	
Unshock Mounted								
Light-Weight	90.9	9.1	22		100.0	0.0	18	
Medium-Weight	50.0	50.0	2		50.0	50.0	2	

Shock mounts quite often present a false sense of security to the designer of shipboard equipment. It is not possible to take any piece of equipment, shock mount it, and expect a high degree of reliability under both shock and vibration; nor can one design for shock alone or vibration alone, forgetting the deleterious effects the other.

A graphic illustration of damages inflicted on equipments tested with and without shock mounts is shown in Figure 2. This graph presents damages resulting from tests on light-weight electronic and nonelectronic equipments, with the damages arranged according to the type of shock test, type of mounting, and the type of mounting adapter used on the shock machine. Furthermore, in order to compare the results directly, the graph has been corrected so that all damages for a particular test are based on 100 equipments. In the lower right-hand corner is given the number of individual tests used in compiling the graph, together with the percentage of the tests which were conducted on electronic units. A number of specifications which required 2-, 3-, and 4-foot hammer drops in the vertical direction instead of 1-, 2-, and 3-foot drops are so indicated. The graph does not show damages to shock-mounted units tested on the IV-A plate adapter since only two or three such units were studied.

Comparing Tests C and D, under which the majority of the more complex equipments were tested, it is observed that the solid-mounted units had fewer damages than those shock mounted. Notice that the percentage of electronic units tested under Test C was approximately 10 percent less than those of Test D, but the absolute quantity of electronic units was greater for Test C than for D.

Because a large percentage of the equipments tested under B were lighting units which must be shock mounted, it would not be fair to compare the damages of Tests A and B as was done for C and D. However, the largest percentage of the damages occurring for all of the tests could have been easily eliminated, regardless of the test or type of mounting, had the units been designed for shock and vibration from the drawing-board stage.

No doubt the logical question which now arises in one's mine is, "Are not the units tested without shock isolators normally more rugged than those tested on shock isolators?". As a partial answer to this question, the electronic units tested under C and D were grouped in 25-pound ranges, with the average number of relays, meters, vacuum tubes, and chassis determined for each range. The minimum and maximum values for the ranges were quite similar in magnitude and would indicate no distinct difference componentwise between equipments tested on either type of mounting. It is believed that the most important reason for the better results shown by solid-mounted units was that designers, knowing their equipments were to be tested

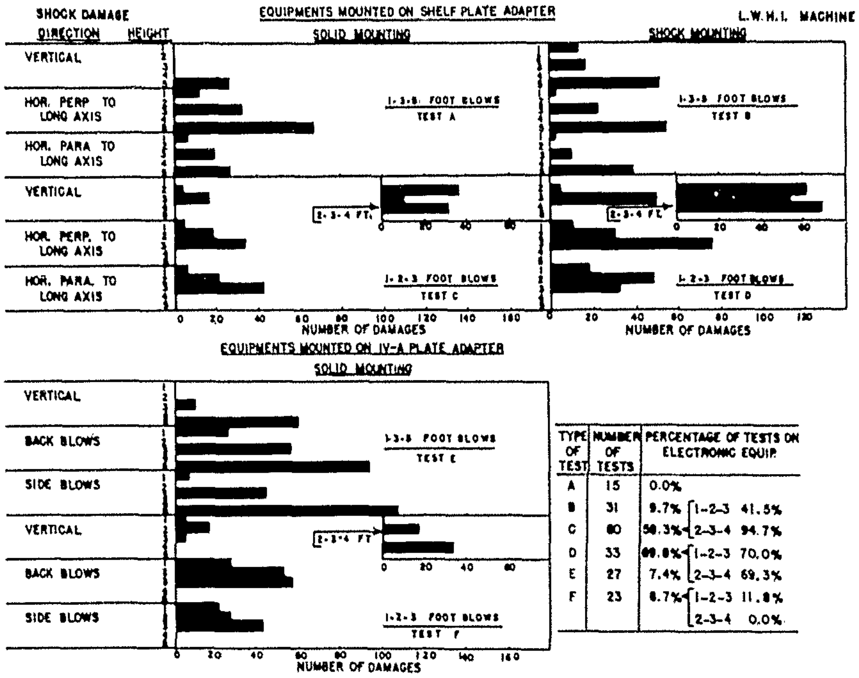


Figure 2 - Damages according to type of mounting, direction, and blow (based on 100 equipments for each type of test)—electronic and nonelectronic equipments

without shock isolators, usually exercised more care in component installation and particular care in construction detail. The greatest percentage of all of the damages studied were relatively minor and could have been eliminated entirely had the designer exercised a little more care in the design of his equipment. Hence, the effectiveness of shock isolators must not be determined on the basis of whether the deflections offered by the mounts during shock afford protection to equipments, but whether or not the average equipment, when designed for damage resistance, is capable of withstanding the tests with or without shock isolators.

Shock mounts definitely are not desirable for ship vibration because of their function as vibration amplifiers, It is seen from this study that the average shock-mounted, light-weight equipment does not appear to be more satisfactory for shock than the unit tested without mounts. Apparently the majority of those damages occurring during shock would have occurred regardless of the method of mounting.

Therefore, for light-weight equipment subjected to 1-, 2-, and 3-foot hammer blows on the Navy's light-weight, high-impact shock machine and to shipboard vibration tests, it appears that solid-mounted equipments, when purposely designed for such tests, produce a higher overall degree of reliability than do shock-mounted equipments. For components which prove themselves to be too fragile for this type of mounting, it might be necessary to individually shock mount the component or to group a number of such components on a separately shock-mounted chassis in the solid-mounted equipment. If more than one chassis required protection, it would probably be better design to shock mount the complete equipment. This procedure would undoubtedly require additional clearance but such clearance does not seem prohibitive in the average unit.

Only five electronic equipments, three of which were shock mounted and two solid mounted, were subjected to 1-, 3-, and 5-foot hammer blows on the light-weight shock machine. Obviously, conclusions cannot be drawn for the higher blows for electronic equipments when only 5 units were studied. But the damages for 1-, 3-, and 5-foot blows to components other than vacuum tubes in nonelectronic equipments were similar to the damages to components in electronic units subjected to 1-, 2-, and 3-foot blows. Electronic equipments subjected to the higher blows would almost without exception require individual or group protection with shock isolators for vacuum tubes and other delicate components; otherwise the comments given for the lower blows would apply.

Because a large majority of medium-weight equipments were tested on mounts, a conclusion on the type of mounting cannot be drawn from experimental results. But the same general types of difficulties were had with medium-weight equipments as with light-weight units, structural damages taking precedence. It seems reasonable to conclude that many medium-weight equipments, when designed for shock and vibration, would produce a higher over-all degree of reliability without mounts than with mounts when individual or group protection is supplied for those components too fragile to be tested solid mounted. Current medium-weight, packaged, electronic equipments usually require shock mounts for the entire system.

There will be certain exceptions to these conclusions. An electronic equipment which has a large portion of its volume filled by a cathode-ray tube would probably be more satisfactory if the complete unit were shock mounted rather than if an attempt were made to mount the tube individually. Likewise, lighting equipment also requires shock mounts, since mounting the complete unit usually is more convenient than supplying protection for the lamps only. There may be a few other isolated cases which would require the entire unit to be shock mounted, but such cases appear to be few.

Generally speaking, the problem of high damage resistance is a mechanical problem. Unfortunately, many of the equipments studied were designed for convenience of electrical circuitry, with little thought given to mechanical performance under the action of inertial forces. Damage resistance is not a characteristic which is achieved by chance, but it is quite real and predictable and can be obtained in part by a knowledge of past damages. Undoubtedly, the future will bring about more damage-resistant equipments because of a closer working relationship between the testing laboratories and the individual designer.

DISCUSSION

Dr. M. G. Scherberg, WADC: Can you tell us what the nature of the damage was on the shock-mounted equipment? What would have been the results of a test comparing equipment designed to be used without shock mounting with shock-mounted equipment?

Woodward: A large percentage of the damages which occurred under shock—that is comparing the damages which occurred to equipment on mounts, and to equipment not on mounts—were quite similar. For example, manufacturers normally design transformers so that the coil and core are supported by a housing. The housing, in turn, has studs on it which fit into the chassis. Under shock the studs sometimes pull out of the housing of the transformer. This occurred for both shock-mounted and unshock-mounted equipment. By attaching the coil and core directly to the mounting bolts, this type of failure is better resisted.

Much is known about stress concentration, but only in a few designs were sharp corners eliminated.

I think that shock mounts provide protection from shock to the equipments. However, we are considering both shock and vibration. We can get the average light-weight piece of equipment through shock tests, very easily, without shock mounts, so why aggravate vibration by placing this equipment on shock mounts? However, in medium-weight equipments we normally have to use shock mounts, particularly on the electronic packaged units.

It may be interesting to point out that about a month ago we shock-tested a packaged electronic equipment weighing about five hundred pounds, without shock mounts, and the repeated damage resolved to only five tubes. Three tubes had a type of clamping system that was not used in the development of the tube; the other tubes had a very heavy plate cap on the glass envelope. We took the caps off and changed the mounting system, and the unit was very satisfactory.

Scherberg: Does that mean the shock-mounted equipments are more vulnerable to vibration?

Woodward: Because of the lower resonant frequencies that you get by mounting equipment on shock mounts, you bring these resonances down to, or at least nearer to testing ranges. Then, the equipment is subjected to higher acceleration forces under vibration. With some of the medium-weight equipments installed on shock mounts, it is very difficult to obtain a resonant frequency above the highest sustained testing frequency. However, we still have the problem of shock, so for the medium-weight equipments we normally have to use shock mounts.

O. J. Stanton, BuOrd: I wonder if there should exist a publication, probably issued by the

Bureaus, which would give the designer of electronic equipment some rules of thumb, or criteria of good mechanical design. In other words, the engineer doing the electronic work would not have to determine where heavy parts should be located, what types of chassis should be used, the designs for chassis construction, and perhaps methods of reinforcing the chassis.

It seems to me that there exist sufficient criteria on chassis design for withstanding shock and vibration, to enable manufacturers to build better equipment.

Woodward: In my thesis which will come out as an NRL report, I have pointed out common errors made by design engineers. For example, many medium-weight equipments are designed with the transformers mounted on the top chassis because of the easier electrical problems involved. However, from our point of view we would like the heavy components mounted low in the equipment in order to raise the natural frequencies.

When considering the location of components on the chassis, an arrangement which places vacuum tubes in the center increases the protection for the tubes because of the larger deflections; the more rugged components can be installed around the chassis edges where they benefit, under vibration, from the stiffer mountings.

However, as far as good structural design is concerned, I feel reasonably safe in saying that in the very near future an NRL report will be forthcoming to point out some of the designs tested in the laboratory and found to be successful.

Stanton: The point I wanted to make was that perhaps it should be issued not by the Shock and Vibration Group, but by the Bureaus as a guide for manufacturers of the equipment. In this way, it will get a much greater distribution than if issued by the Shock and Vibration Group.

Woodward: Your point is well taken.

K. G. Moeller, NEES Annapolis: I want to make a comment on Naval practice in mounting equipment. I presume it is not true any more that the Navy generally mounts its equipments on 20-cycle natural-frequency mounts, because the noise requirements are not satisfied by this type of mount. We are going down now to five or six cycles. Since the development, at our station, of a mount which has nonlinear characteristics so that it does not require such large excursions, five- or six-cycle mounts can be used in protecting equipment for shock only.

One statement you made is not clear to me; you say that in order to have the equipment pass the test, you have to make certain changes. The main reason for mounting equipment is to provide stability aboard ship; if the test does not verify this condition, then we should change the test.

Woodward: We hope the testing specifications simulate shipboard conditions.

Moeller: They do, to a certain extent, but low-frequency mounts would help in noise reduction and shock protection. Other design rules must be followed, of course. For instance, you can put your mounts as close to the center of gravity as possible. The assembly of several items on a common base is helpful because the total mounted mass is greater and the acceleration smaller.

I would like to mention a black book issued by the Bureau of Ships about four years ago, called "Shock on Naval Ships." Then there is a red book which contains some good design rules for improval of naval equipment, but is not too well known.

Dr. W. H. Hoppmann, Johns Hopkins U.: I am in the academic field. Would you care to say what thesis your paper is based on?

Woodward: In the Naval Research Laboratory we have a scientific training program which works in cooperation with the University of Maryland Graduate School. I was granted permission to write a thesis based upon the work we are doing, i.e., design and evaluation tests performed by the Shock and Vibration Group of the Laboratory.

Dr. I. Vigness, NRL: Here is a hint for supervisors. If you want to get some work done inexpensively, i.e., on the job supplemented by a lot of homework, have an educational program such as was just mentioned. The employee works on a thesis dealing with something that is worthwhile to the government. He does a great deal of homework and spends a little time in the laboratory. In this way, you get results that you would not get otherwise.

Dr. Wensel: In regard to the question of whether you are building equipment for testing passage or shipboard passage, I should think that after you procure a certain amount of this material you might finally, on the basis of these tests, get to a point where some correlation appears. You get to where you are buying what you want.

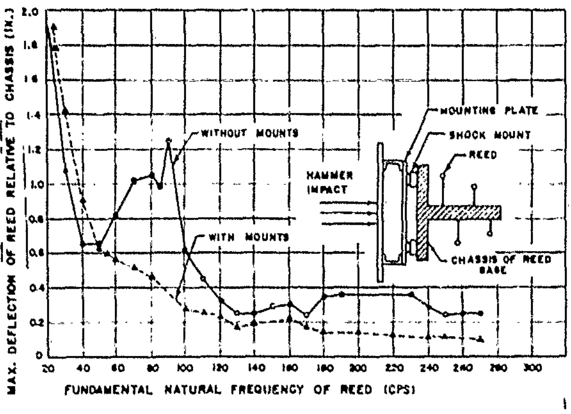


Figure 3 - Deflection data from multifrequency reed gage (five-ft hammer drop on high-impact shock testing machine for light-weight equipment)

C. E. Crede, Barry Corp.: Mr. Woodward's paper sets forth, among other things, data on failures of equipment components during shock tests. The data are broken down to show the number of failures that occurred when shock mounts were used, and the number that occurred when shock mounts were not used. Such comparisons are not conclusive unless the results are examined statistically. It is possible that equipment supported by shock mounts experiences greater damage than similar equipment rigidly mounted, if the design of the equipment is not such as to take advantage of the protection afforded by the mounts.

The nature of failure as a result of shock may be illustrated by reference to a hypothetical equipment. This hypothetical or typical equipment consists of many structures, each with its characteristic natural frequency; the equipment may be simulated by a device that has become known as a multifrequency reed gage. Such a gage is made up of a chassis, a number of reeds of different natural frequencies, and means to indicate the maximum deflection of each reed during shock. If the deflections of similar reeds are compared for different conditions of shock, the relative severity of the respective shocks will be indicated by the relative deflections of similar reeds.

Shock tests have been made on the Navy high-impact shock-testing machine, first with a reed gage mounted rigidly to the mounting plate of the testing machine, and then with the gage attached to the mounting plate by shock mounts. By varying the natural frequencies of the reeds, the effect of the shock upon structures of various natural frequencies can be determined. The results are shown in Figures 3 and 4 which are prints of Barry Corporation Data Sheet No. 150. Figure 3 shows the maximum deflection of the reeds, and Figure 4 shows the maximum acceleration of the same reeds, both expressed as a function of the fundamental natural frequency of the reed.

The amplitudes of the curves on the two sheets indicate the relative severity of the shock experienced by the equipment. It is apparent that the solid curves, which apply to tests without mounts, show relatively high peaks at approximately 90 cps. These peaks occur because the mounting plate of the shock-testing machine has a natural frequency of 90 cps, and the reeds experienced large amplitudes as a result of resonance between the reeds and the mounting plate. The dotted curves, representing tests with mounts, show little evidence of the peak at 90 cps. The mounts thus function as vibration isolators to isolate the transient vibration of the plate resulting from the hammer impact of the shock machine.

It will be noted that the dotted curve is above the solid curve for natural frequencies less than 50 cps. This higher level is the result of resonance between the various reeds and the shock mounts. There is a peak having much in common with the peak that occurs in the solid curve at 90 cps as a result of resonance between the reed and the mounting plate. Therefore, it is evident that elements with natural frequencies less than 50 cps would be less likely to experience damage if not supported by shock mounts. Elements with natural frequencies greater than 50 cps tend to receive protection from the mounts. It is therefore quite logical that more failures would be experienced in shock-mounted equipment if the structures of the equipment tend to be flexible. Equipment whose structures tend to be rigid is less susceptible to damage if shock mounted.

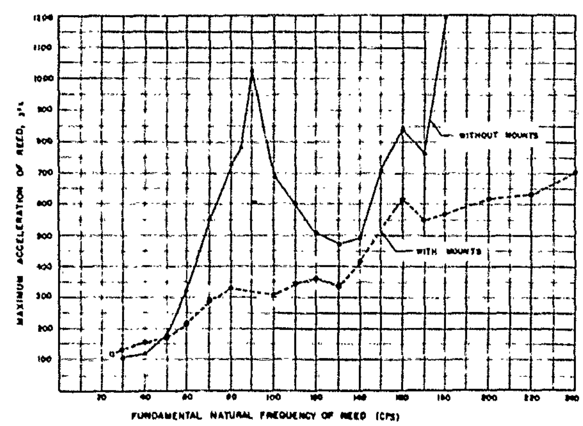


Figure 4 - Response spectra from multifrequency reed gage (five-ft hammer drop on high-impact testing machine for light-weight equipment)

The mounting plate of the shock machine may be considered to simulate a particular type of mounting for the equipment. When the equipment is mounted in a naval vessel, it will probably be mounted on a bracket whose natural frequency is different from the natural frequency of the mounting plate on the shock machine. The resonant condition will then occur at another frequency. A structure of the equipment that did not experience such resonance during the shock test may then tend to experience resonant conditions when installed within the vessel. The shock test thus tends to become invalid in a sense, because it fails to predict the difficulty that may occur as a result of resonance between the structures of equipments and the mounting brackets.

It is fortunate that the use of mounts almost completely eliminates the effect of these resonances and creates a condition in which the natural frequencies of the mounting brackets are unimportant. The natural frequencies of mounting brackets are difficult to predict and almost impossible to control. If reliance is to be placed on the validity of shock test, it will help if shock mounts are used with all equipments because they tend to eliminate inconsistencies between shock-test conditions and actual service conditions.

Two principal functions for shock mounts may, therefore be outlined as follows:

a. The shock mounts are responsible for a lower stress in elements of the equipment that are stiff enough to take advantage of the protection afforded by the mounts. It is evident from Figures 3 and 4, that the protection afforded by the mounts is relatively modest in any instance, and that mounts cannot create rugged equipment from equipment that lacks ability to withstand shock tests.

b. A second function of shock mounts is to eliminate inconsistencies between shock tests and service conditions. Such tests tend to become more reliable in the sense that the importance of peculiarities of the test is minimized when equipment is subjected to shock tests for qualification purposes.

Woodward: I do not believe there is any argument relative to the comments by Mr. Crede. In fact most of the remarks in regard to protection offered by mounts for the high-frequency components of shock were briefly summarized in a statement in the second paragraph of the paper. Referring to the low-frequency shock components, it is a very difficult thing from a practical engineering standpoint to design a cabinet which has a natural frequency above 50 cps. This frequency usually can be reached in the smaller light-weight equipments, but in the medium-weight units, 35 to 40 cps appears to be the general limiting value. Shock, however, is only one phase of the problem as it concerns the Navy; vibration constitutes the other phase even though it does not receive sufficient consideration in time to do much about it.

The primary purpose of the paper is to acquaint designers with the difficulties of designing for shock and vibration. Real equipments and very real damages were examined in the process. The statements made with reference to shock mounts include the damaging effects of both types of tests and are usually conditional. It is not felt that the manner in which the results are presented is the best; but it serves to give the individual who is actually doing design work a better and more complete over-all picture of the problem. It is hoped that more thinking will be directed toward shock and vibration, not simply toward shock alone.

* * *

FORCES ACTING ON A SHIPPING CONTAINER IN TRANSIT

K. W. Johnson, WADC

Presented by C. A. Golueke, WADC

Data are presented on rail, air, and handling shock measured under representative service conditions. Considerations in establishing laboratory tests from field data are briefly discussed. The discussion includes the work to date in the establishing of vibration and shock design criteria for airborne equipment.

This paper gives information and data, on measured vibration and shock experienced in rail transportation and in handling, accumulated since the 16th Symposium held in Chicago. At that symposium, vibration data were presented on aircraft conditions as a result of work by the Curtiss-Wright Corporation. Since then, the project entitled "Vibration and Shock Design Criteria for Airborne Electronic Equipment" has been progressing satisfactorily. The first phase—finding the disturbance under service conditions and defining the envelope of the disturbance—has been completed by Curtiss-Wright and by North American Aviation, Inc. The second phase, having to do with establishing realistic vibration and shock test procedures and determining what equipment is required, is presently under contract with the Barry Corporation. The third phase of the work, which is the determination of the design level of present equipments with regard to vibration and shock, is under contract with the Armour Research Foundation. The progress of each of the phases will be discussed briefly here.

MEASURING SERVICE CONDITIONS

In order to make this data more complete, two graphs on aircraft conditions, presented at the 16th Symposium, are included. Figure 1 shows data on flight conditions for aircraft with reciprocating engines. Figure 2 shows vibratory conditions in aircraft powered by jet engines. It is noted that the vibrations experienced in jet-type aircraft are no less severe, and in some cases more severe, than those in the reciprocating type.

Figure 3 shows shock conditions in the aircraft fuselage section where most of the electronic equipment is located. The values are the results of carrier landings and take-offs. The intensity ranges from a few g's to 17 g's, and the durations, from 7 to 14 ms. Figure 4 shows shock experienced in the wing section. Intensities to 38 g's were recorded, with duration ranging generally from 22 to 80 ms. The larger values were mainly in the 22- to 40-ms range. Such durations are to be expected in these two areas since the structures of aircraft are relatively elastic.

Figure 5 shows conditions measured while bumping freight cars during switching operation. These measurements were made in cooperation with the Southern Pacific Railroad. The measuring equipment used for the tests consisted of a Miller Type-H twelve-channel recording oscillograph, with Calidyne accelerometers. The test cars were Class "A" of riveted construction, one approximately two years old, called the new car, and one 15 years old, called the old car. Various combinations of

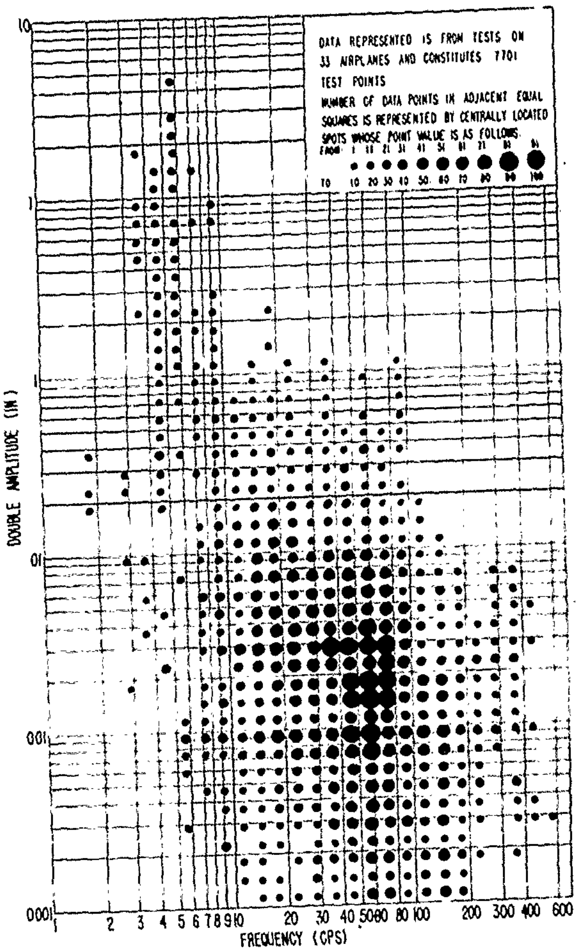


Figure 1 - Reciprocating engine aircraft—flight condition

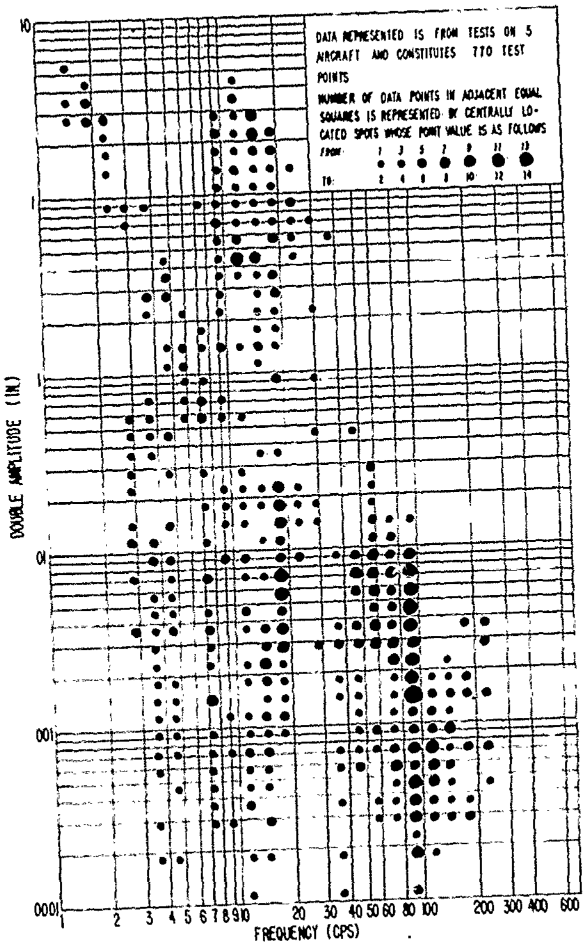


Figure 2 - Jet engine aircraft—flight condition

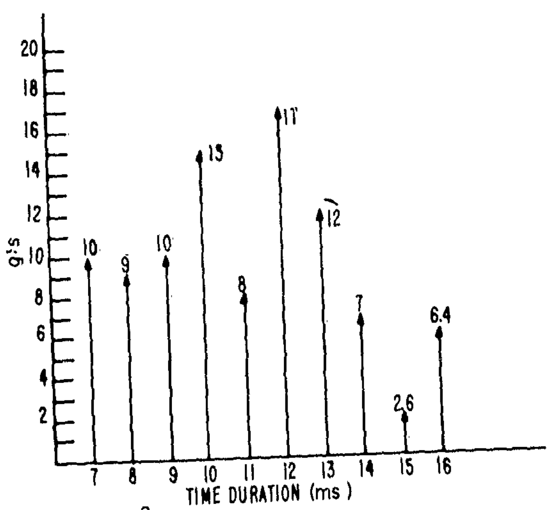


Figure 3 - Shock in fuselage section of aircraft—carrier landing and take-off

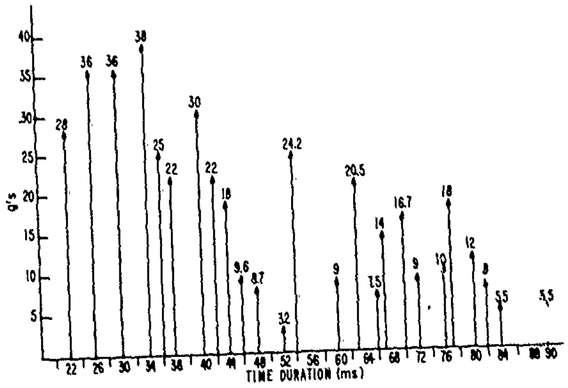


Figure 4 - Shock in wing section of aircraft—carrier landing and take-off

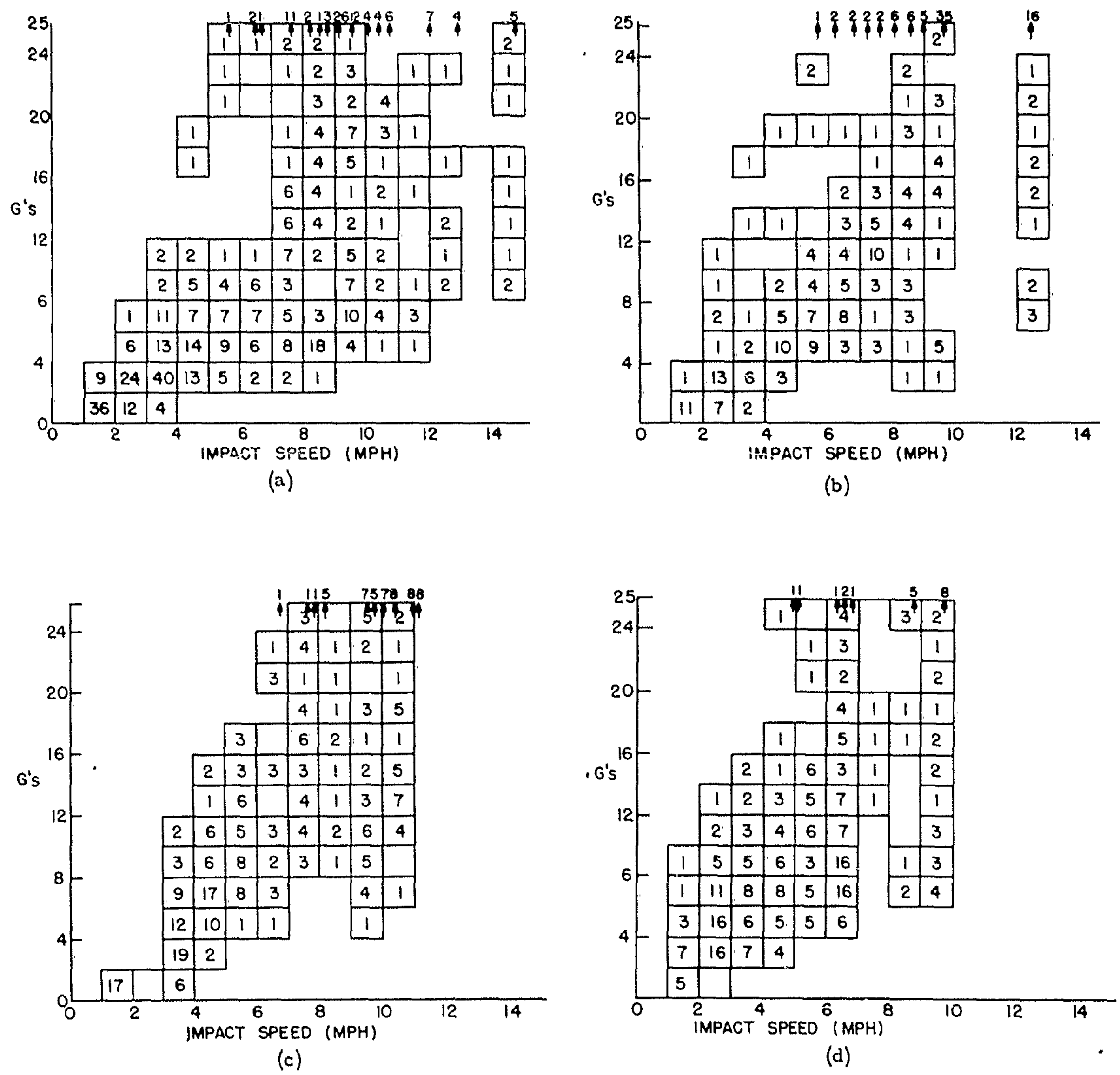


Figure 5 - Shock conditions measured on freight cars during switching operation

(a) Test series 2 - New car, light load (18,000 lb)
(b) Test series 3 - New car, heavy load (36,000 lb)
(c) Test series 6 - Old car, light load
(d) Test series 10 - Old car, light load

light and heavy loading with various type draft gear were employed. Electronic equipment always is transported in Class "A" cars. In all test series except one, the instrumented car was impacted into a string of six loaded and ballast cars. On impact, the accelerometers in channels 2, 4, 5, and 6 tended to vibrate at their normal natural frequencies—in the order of 400 cps. In order that the records containing those spurious high-frequency components might be read, allowances had to be made for the effect of these components on the actual galvanometer deflections. By comparing these values with those from accelerometers not showing the above tendency, the error was found to be slight. Values in excess of 25 g's were recorded with rated 25-g pickups. These values are shown by an arrow at the top of the figures at the 25-g level. Speeds of impact range from 1 to 14.6 mph.

Figure 6 is a summation density plot of all bumping tests on both old and new cars with heavy and light loads. Duration of shock in these tests ranges from 3 ms to 95 ms with a majority of shocks lasting from 10 to 20 ms.

The ground handling tests were conducted using as the stopping media hard earth, wood, and concrete. The heights of drop were 14, 28, and 42 in. Table 1 gives the values for 7-3/4-lb electronic equipment unpackaged. For each

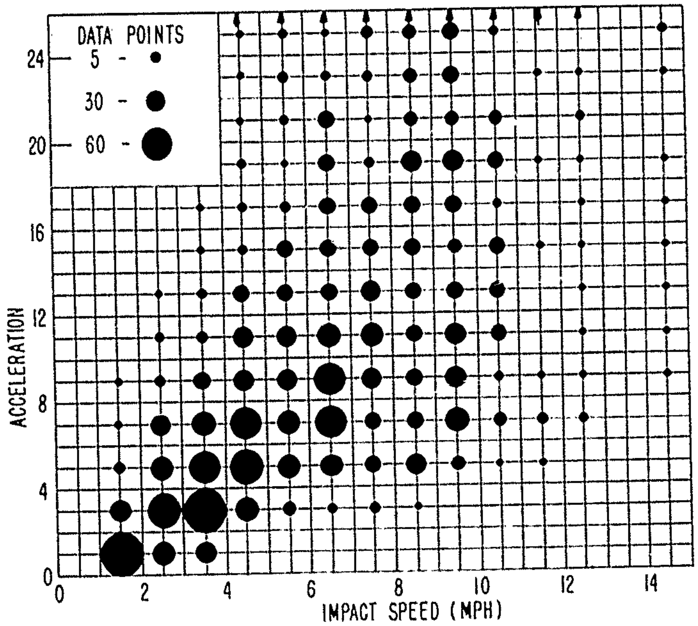


Figure 6 – Acceleration density plot of all bumping tests – new and old cars, light and heavy loads

TABLE 1

GROUND HANDLING DATA
Unpackaged 7-3/4-lb electronic equipment

HEIGHT (in.)	FACE	TYPE FLOOR	DIRECTION					
			VERT.		LAT.		LONG.	
			g's	TIME (ms)	g's	TIME (ms)	g's	TIME (ms)
14	CORNER	EARTH	85	0.85	31	5.00	38.4	4.65
14	BOTTOM	"	—	—	—	—	—	—
14	CORNER	WOOD	170	1.05	235	3.4	79	2.25
14	BOTTOM	"	199	0.85	188	1.25	71.7	2.50
14	CORNER	CONCRETE	182	1.25	261	1.35	263	1.90
14	BOTTOM	"	400	1.08	460	0.74	267	0.675
28	CORNER	EARTH	57.5	1.40	26	1.20	130	5.00
28	BOTTOM	"	287	2.00	312	0.80	247	0.75
28	CORNER	WOOD	115	2.50	52	1.00	123	3.00
28	BOTTOM	"	402	1.40	364	0.80	205	0.85
28	CORNER	CONCRETE	690	1.05	364	0.75	328	1.00
28	BOTTOM	"	690	1.10	364	0.80	288	0.75
42	CORNER	EARTH	223	0.69	208	1.25	173	2.3
42	BOTTOM	"	—	—	—	—	—	—
42	CORNER	WOOD	211	1.2	104	1.04	164	3.80
42	BOTTOM	"	534	1.1	55.4	1.05	247	0.65
42	CORNER	CONCRETE	169	1.55	260	1.93	164	2.86
42	BOTTOM	"	463	1.55	312	0.90	164	0.55

height of drop are shown the intensities and durations for the corner and bottom faces. Three directions were recorded. Other faces and edges were measured, but for the purpose of this paper, it was felt that the corner and bottom face were representative. Intensities range from 115 g's to 690 g's in the vertical direction. The lateral direction shows values from 26 g's to 460 g's. In the longitudinal direction, 38 to 228 g's were recorded. Time durations are in the 1- to 4-ms range. Table 2 is for a 33-lb load. Intensities vary similarly to those in the preceding table. It is noted that the durations become longer with the increase in weight.

Table 3 presents data for a load of 65 lb. Intensities are generally less with greater durations, which means that with the heavier load the outside structure is yielding more. Figure 7 is a density plot of all drop tests. Note that the "g" level does not increase to any great extent with increase in height of drop.

Table 4 presents data on an equipment within a shipping container. The container was made of pine with a moisture-impervious barrier inside of which was a corrugated fiberboard box. Next, there were two layers of corrugated fiberboard filler with a metal foil box around the equipment. This package conformed to Specification JAN-P-13A, Method II. Values are shown that were recorded on the equipment and on the container. It is noted that the equipment values are much less than the container values, indicating that the cushioning for this particular load was very good. In the vertical direction intensities as high as 662 g's were attenuated to 38 g's. The reduction was not so great in the longitudinal direction. Figure 8 is a density plot of the packaged equipment in comparison to the "g" value on the outside of the package.

Although the functional aspects of the equipments were not considered in the tests, some visual observations were made which are interesting. The cases of all unpackaged units were badly distorted, and crushing of the corners occurred. The damage to components was more severe in the lighter units than in the heavy units. A 5-in. cathode-ray tube in the 65-lb piece of equipment withstood the entire series of drops without visible damage. In the 7-3/4-lb unit, ceramic coil forms were shattered and most of the vacuum tubes were destroyed. The packaged unit showed no damage at all.

It is pointed out that the data presented in this paper are only representative of the vast amount of data accumulated under the contract. Further detailed information is contained in the reference.

TABLE 2

GROUND HANDLING DATA
33-lb equipment

HEIGHT (in.)	FACE	TYPE FLOOR	DIRECTION					
			VERT.		LAT.		LONG.	
			g's	TIME (ms)	g's	TIME (ms)	g's	TIME (ms)
14	CORNER	HARD EARTH	80	1.73	88	1.12	270	3.62
14	BOTTOM	"	187	1.08	264	0.80	110	3.70
14	CORNER	WOOD	80	1.25	201	1.14	480	1.02
14	BOTTOM	"	133	1.48	151	0.57	117	4.72
14	CORNER	CONCRETE	133	1.36	301	0.41	585	0.85
14	BOTTOM	"	200	1.79	288	0.53	193	0.72
28	CORNER	HARD EARTH	133	0.88	239	0.61	575	0.94
28	BOTTOM	"	560	2.76	226	1.30	233	1.25
28	CORNER	WOOD	80	1.25	63	1.62	154	1.12
28	BOTTOM	"	220	1.48	251	0.57	205	0.68
28	CORNER	CONCRETE	244	0.83	313	0.83	274	0.83
28	BOTTOM	"	396	1.31	OFF SCALE	—	292	0.95
42	CORNER	HARD EARTH	103	5.45	448	0.83	164	3.03
42	BOTTOM	"	192	3.75	309	1.00	247	1.12
42	CORNER	WOOD	144	1.56	243	0.62	99	8.75
42	BOTTOM	"	240	1.00	265	0.64	329	0.71
42	CORNER	CONCRETE	167	0.54	294	0.54	106	7.15
42	BOTTOM	"	467	2.42	695	2.00	443	0.75

In summary, it is felt that the data presented may be used to form a basis for establishing vibration and shock tests for future military specifications.

TABLE 3

GROUND HANDLING DATA
65-lb equipment

HEIGHT (in.)	FACE	TYPE FLOOR	DIRECTION					
			VERT.		LAT.		LONG.	
			g's	TIME (ms)	g's	TIME (ms)	g's	TIME (ms)
14	CORNER	HARD EARTH	20	3.50	41	1.00	58	0.75
14	BOTTOM	"	—	—	—	—	—	—
14	CORNER	WOOD	40	11.70	27	1.93	41	5.34
14	BOTTOM	"	120	1.96	95	0.82	82	0.82
14	CORNER	CONCRETE	61	1.39	98	1.17	125	0.72
14	BOTTOM	"	213	2.56	137	0.78	158	0.50
28	CORNER	HARD EARTH	122	2.72	59	0.76	110	3.97
28	BOTTOM	"	122	1.76	59	0.85	78	0.90
28	CORNER	WOOD	76	1.53	90	0.79	84	0.96
28	BOTTOM	"	—	—	—	—	—	—
28	CORNER	CONCRETE	—	—	—	—	—	—
28	BOTTOM	"	381	2.61	453	0.99	286	2.15
42	CORNER	HARD EARTH	137	8.04	587	33.50	439	25.75
42	BOTTOM	"	260	3.40	163	0.49	219	1.39
42	CORNER	WOOD	79	12.67	224	45.00	125	15.64
42	BOTTOM	"	269	3.95	470	0.82	253	1.22
42	CORNER	CONCRETE	134	9.56	506	22.8	423	24.00
42	BOTTOM	"	323	3.92	269	1.82	524	1.01

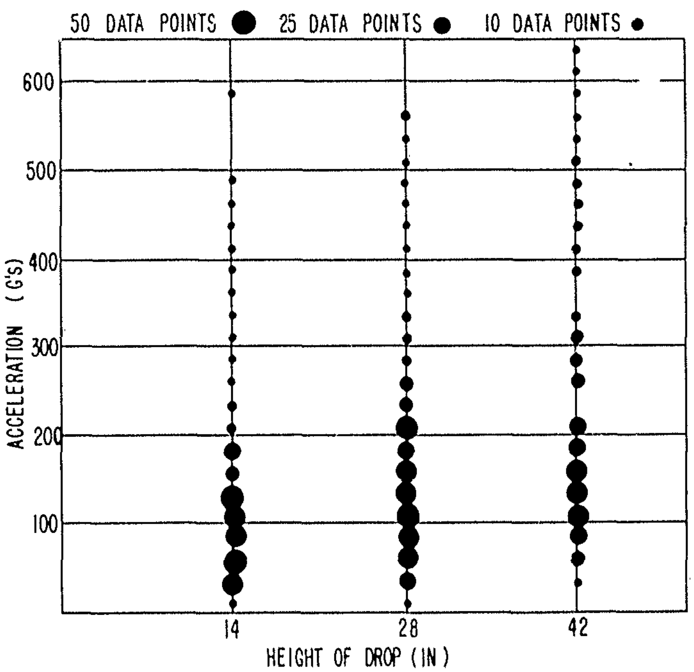


Figure 7 - Density plot of all drop tests on unpackaged electronic equipments

TABLE 4

GROUND HANDLING DATA

Equipment within case - total weight 73 lb

HEIGHT (in.)	FACE	TYPE FLOOR	DIRECTION					
			VERT.		LAT.		LONG.	
			g's	TIME (ms)	g's	TIME (ms)	g's	TIME (ms)
14	CORNER	HARD EARTH	C 103	2.62	—	—	69	2.32
			E 42	13.50	38	4.00	63	2.26
14	BOTTOM	"	C 310	2.20	289	9.80	110	1.13
			E 33	80.00	31	4.17	31	4.29
14	CORNER	WOOD	C 78	4.05	—	—	41	.89
			E 23	55.00	31	8.16	57	2.27
14	BOTTOM	"	C 310	1.02	123	0.68	55	1.88
			E 18	17.00	13	3.63	31	2.16
14	CORNER	CONCRETE	C 207	2.22	—	—	82	0.76
			E 41	10.28	76	2.88	57	1.69
14	BOTTOM	"	C 207	0.17	103	2.44	137	0.57
			E 23	8.81	19	3.98	19	1.59
28	CORNER	HARD EARTH	C 310	0.32	103	3.20	69	0.32
			E 36	35.00	44	8.02	31	2.50
28	BOTTOM	"	C 568	0.54	—	—	178	1.07
			E 53	12.67	57	1.19	50	2.78
28	CORNER	WOOD	C 310	0.17	82	2.16	68	0.33
			E 47	11.40	70	12.68	31	2.75
28	BOTTOM	"	C 491	0.32	144	0.58	123	0.39
			E 35	12.84	44	6.03	19	2.18
28	CORNER	CONCRETE	C —	—	—	—	—	—
			E —	—	—	—	—	—
28	BOTTOM	"	C 662	3.88	219	0.37	150	0.37
			E 38	7.35	49	3.40	32	1.76

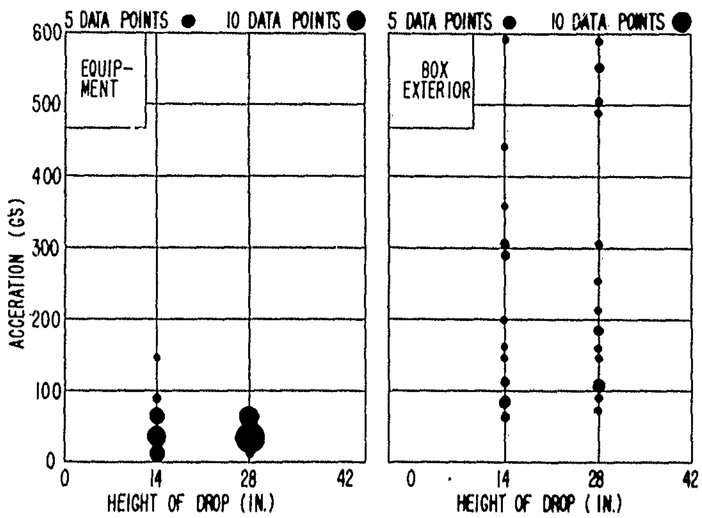


Figure 8 – Density plot of acceleration in packaging (JAN-P-106A)

REFERENCE

Riedhart, W. H., and Gulcher, R. H., Final engineering report on investigation of vibration and shock requirements for airborne electronic equipment, North American Aviation Report No. 120X-7 (Restricted), 1,485 pp, March 30, 1951.

DISCUSSION

ESTABLISHING TEST PROCEDURES
(Discussion paper submitted by K. W. Johnson, WADC)

Today, unrealistic test procedures waste more time, money, and materials than any other single set of conditions. Many of the tests which are currently in use were originally right in principle when initiated for a specific item. However, through the years, with changing personnel, the tests have been adapted blindly for conditions other than the original conditions—conditions which involve different modes of transportation—and the original meaning of the test has been lost. As a result of the misapplication of vibration and shock tests, in order to meet specifications, equipments have been over- and under-designed for actual service conditions.

In view of these facts, the importance of realistic testing cannot be overemphasized. The project engineer should bear in mind at all times that tests influence all items from the very beginning of research, through the development and manufacturing stages, and into the actual operation. The engineer's job is not completed merely by selecting a standard vibration and shock test from some general specification and then incorporating this test in his detailed specifications. He must consider the extent to which the selected standard test meets the requirements of his item, the operating of the item for test, and the determination of the test validity.

To illustrate the importance of selecting a standard test which meets the requirements of an item, let us assume that the item is test equipment which is mainly used on the ground but which must be air transportable. The standard test for the item is based upon operation under flight conditions. The range of frequencies and vibration amplitudes of the item to be tested may be the same as the range established in the standard test. However, the minimum of test time should be less than that called for in the standard specification since the test equipment's time under flight is less.

Positioning the item for test is another important consideration. The general specification usually states "...in a manner as normally used in service..." Noncompliance with this requirement can change entirely the value of the test results. The component tests are the worst offenders of the positioning requirement. For normal service use, many components are used in series with other components or they are held in place by wire leads. Instead of testing them realistically, the component in many instances is tested in a rigid jig which provides no similarity to the actual service condition. Such tests are comparative and must be interpreted as such. Extreme caution must be exercised in the evaluation of a comparative test where the component combinations have been eliminated. The danger of misinterpretation is obvious since it is not generally known what effect the eliminated component has on the test item. Such tests usually lead to requirements which are far in excess of the requirements of those which are practical and necessary for one of the linkage components. This wastes engineering time and money.

The project engineer must consider and include the determination of the test's validity. In other words, he must be sure that the basic principle of the test is applicable to his particular item. The practice of establishing detailed vibration and shock requirements by merely selecting a standard test paragraph—without investigating the actual meaning of the test—is much too common. This erroneous practice is especially prevalent when shock tests are being selected. To many electronics engineers, all shock tests are alike, whereas, actually, two shock tests may be completely different.

In selecting the applicable shock test, the engineer should first consider the degree of similarity between the vibratory test and the service condition. Such questions as the following need to be answered:

1. In service, are the forces which result from shock impact applied directly to the internal structure and components, or do

the forces first cause damage to the outer case and, in turn, are they transmitted to the internal structure?

2. Are the forces constant for a period of time or are they transient?

3. Are the shocks of long or short duration, and what level of "g" intensity is present?

If the forces are applied directly to the internal structure, a test may be used that requires the test item to be attached to a shock machine or to a carriage. The table makes the actual contact with the force-generating medium. If the case or the dust cover makes the actual contact first, a machine type of test, as indicated in the preceding information, is not representative and a random drop test of the item onto a hard surface is more realistic. If the forces are constant for a period of one second or longer, a centrifuge type of test should be used. For a single-impact-transient type of shock, a drop or hammer shock machine is generally used. The time duration will establish the range of the pulse desired as well as give an indication of the amount of energy present if the "g" level is known. Knowledge of all of these factors will aid in determining whether or not the basic principle of the selected test is satisfactory.

Once a realistic test is established in principle, it is necessary to determine the procedure that applies to the end purpose of the test. The two primary purposes of vibration and shock tests are to evaluate the item for fatigue and for ultimate strength. Generally, vibration tests are associated with fatigue, and shock tests with ultimate strength. Such separation is erroneous, since both purposes should be considered under each condition. At present, many of the tests specified are confusing because these two purposes have not been defined in the test procedure. Long-term vibration tests (fatigue) are being specified with maximum drop amplitude values (ultimate strength) and a large number of shock impulses (fatigue), with maximum "g" intensity (ultimate strength).

Vibration tests may be accomplished either by cycling an item over a frequency range or by cycling it at the item's resonance frequency. Since the point of resonance is the most damaging, the resonance tests will give quicker results. However, the resonance test is practical only with a one-mass system in a single-degree-of-freedom system. The mass of an electronic equipment, for instance, is made up of several hundred small component masses with each small mass having its own point of resonance. It is impossible to locate all of these resonance points. Therefore, the cycling test is used so that each component will be at resonance during a percentage of the time.

The procedures for the vibration fatigue test for equipments used with vibration isolators are different from those for equipments which do not have isolators. For equipments with isolators, the frequency range need not extend beyond 55 cps. The reason for this limited frequency range is that the predominant resonances of the chassis, holding brackets, and components are generally below 55 cps, and for those that occur above this frequency, the attenuation of vibration caused by the isolators is between 95 percent and 99 percent. For equipment without isolators, the frequency range should extend to at least 500 cps. Without isolators, there is no control of the aircraft structure resonances at the equipment attaching point.

The amplitude of vibrations for the fatigue test should be representative of the average steady-state service condition. The fatigue test should be conducted for a period of time equal to the expected life of the item. In most instances, such a test is not practical for an equipment or component at the time of first article test. However, it is possible for the manufacturer to conduct such tests prior to or after the first article test and thereby completely evaluate the design. Such long-term life tests are common for civilian items. It seems even more important that such tests be applied to military items where so much more is at stake. Since the component manufacturer should know more about his component than anyone else, it is logical that the manufacturer should conduct fatigue tests. The project engineer should request such tests when establishing testing specifications.

The vibratory ultimate-strength test should be for a short period of time. Maximum double-amplitude values of the service condition should be used.

The shock-fatigue test should be established on low-value shock (2 or 3 g's) of the type that the equipment experiences daily. The test should be stated in terms of hours rather than in terms of the number of shocks. The shock-fatigue test is vibratory in nature because it is actuated by a shock impulse which excites the individual resonance of component structures simultaneously. The test is different from the

normal vibratory-resonance tests in that the motion of the resonance structure peaks at the time of impulse and reduces to zero on a decaying-resonance curve, the motion being repeated for a specified number of shock impulses or in terms of hours at a specific rate. This test is very realistic and a close correlation with service conditions may be achieved.

The shock ultimate strength test should be conducted with a limited number of shocks. The "g"-level intensity should be based upon the maximum number of g's experienced in service. It must be remembered that this test is for ultimate strength only, and, as such, should be limited to a maximum of 10 or 12 shocks; otherwise, the damage will be a result of the cumulative effect of the shock rather than impact loading. In establishing the maximum "g" level, it is not advisable to double or triple a service-level "g" for the purpose of a safety factor. A shock is the conversion of potential energy into strain energy in the structure of the equipment and, if excessive intensities are used, the realistic value of the test will be lost. A sufficient safety factor is achieved in the number of repeated shocks.

(Discussion presented by C. E. Crede, Barry Corp.)

The problem of specifying laboratory tests to qualify equipment for actual service conditions has both steady-state and transient aspects. The approaches to these two different parts of the problem are quite different, and parallel programs are being followed.

The steady-state problem makes use of vibration data collected from many sources and correlated by North American Aviation. The data are presented as points on a chart with vibration frequency as one coordinate and vibration amplitude as the other. A curve of maximum anticipated vibration amplitude, as a function of frequency, may then be drawn to envelop the plotted points. This envelope can be interpreted in terms of the acceleration level that the equipment must withstand for a very large number of cycles during actual service without failing. These vibration data can then be correlated with experimentally established curves, giving the endurance properties of equipment and components. Such properties may be expressed in terms of number of cycles to failure, as a function of maximum acceleration embodied in the test. This corresponds to the conventional stress-cycle curve used to define the endurance property of a material. It is an objective of the program to use the acceleration-vs-cycles-to-failure curve to determine the acceleration increase necessary to cause the same failure in the laboratory, after relatively few cycles, as would occur in service after many cycles. Available acceleration-cycle curves for electronic components are preliminary and tentative. This phase of the investigation has not been carried far enough to indicate the extent to which acceleration-cycle curves, as finally obtained for electronic components, can be used to correlate laboratory and service failures.

The transient aspect of the problem is also related to the acceleration-cycle curves, but in a somewhat different way. Acceleration-cycle curves are conventionally obtained by repeated stressing of the material being tested, the test being conducted in such a way that the stress amplitude reaches the same maximum at each cycle of the test. In a transient excitation, the maximum stress is different at each cycle until the transient vibration dies out. The relation between endurance tests at constant maximum stress and fatigue tests at variable maximum stress has been investigated by Miner and others. It is expected that investigations conducted along these lines will provide a guide for the use of the acceleration-cycle curves in evaluating the severity of the transient conditions.

The technique being used in investigating the transient problem employs a specially designed function generator which will convert an oscillogram into an electrical signal and an electronic analog of a mechanical system. The input to an analog computer is acceleration as a function of time, as measured on aircraft during conditions of shock or transient vibration. The output of the analog computer represents the response to this input of elastic systems of various natural frequencies. Such data may then be used to calculate a damage coefficient, using the principles set forth by Miner. It is then proposed to use, as a further input to the analog computer, acceleration as a function of time, as measured on shock-testing machines. A damage coefficient for the testing machine can then be calculated as above. It is hoped in this way to determine the necessary characteristics of a shock pulse that will produce the same damage coefficient as service conditions, considering the smaller number of applications of shock in the laboratory as compared with the number encountered in actual service.

DETERMINING DESIGN LEVEL OF PRESENT EQUIPMENTS

(Discussion presented by Fred Mintz, Armour Research Foundation)

It is hoped that the testing of components under specified test procedures, to evaluate their ability to survive under known field disturbances, will lead finally to sufficient information which will allow the preparation of a design manual—the fourth and final phase of the project.

There are three possible approaches to this testing program. One is to define the behavior of specific components under vibrations in general—say, in terms of response versus frequency—and then to compute the response to the known vibration spectrum envelope as determined from service conditions. This assumes, among other things, linearity and simplicity of the system. For many of the components in use, it has been demonstrated that the system is neither linear nor simple.

The second approach is to observe the response of a number of these items to a specified test procedure which logically represents the maximum disturbance expected to be encountered in field service. This disturbance might be termed the design level for survivability. It probably represents the envelope of the maximum observed vibration in terms of amplitude versus frequency.

The third approach involves accelerated test procedures. This requires the design and application of suitable test procedures that will enable us to determine in a relatively short time the survivability of the equipment under fairly long-term excitation of the severity encountered in service.

The concept of a three-dimensional surface has been found useful in considering the simplification of test procedures. The parameters of this surface are $\underline{n}$, the number of cycles; $\underline{a}$, the acceleration level required to produce failure; and $\underline{f}$, the frequency. It requires a great many data points to establish such a surface (which might be called the NAF surface).

The first way to decrease the number of data points is to work at resonance. Certainly we know that in general a system subjected to a given amplitude of vibration at resonance is more likely to fail than if it is vibrated off resonance. Consequently, the first thing to do is to find the resonance curve, i.e., the response-versus-frequency characteristic of the system. Then, you study each component and attempt to define the most critical resonance in order to work with only one frequency, thereby decreasing the number of variables. If you are working at resonant frequency, and if you obtain fatigue-type failures, you can, by testing several samples at a given level, specify a number that represents the number of cycles to failure.

With many of the components that we have tested, particularly tubes, this cannot be done. There is a random distribution of failures, some of which occur with little or no vibration applied. This seems to demonstrate the need for some quality control at the tube manufacturing plants so that you are not confronted with an appreciable supply of tubes off the shelves that fail to function properly when taken out of the box.

A curve actually can be plotted to give, not the number of cycles to failure, but rather the relation between number of items surviving the number of cycles of vibration applied. If you can find some relation between the acceleration level applied and a significant parameter (e.g., slope) of this curve, then you can get a curve (which should be essentially monotonic) that defines the level of the excitation versus the variation of the significant parameter of the earlier curve.

In this way we hope to be able to define the behavior of the component. Thus, we should be able to establish relative survivability on the basis of the accelerated test and hope to establish some sort of approximate absolute survivability. We shall need to know the maximum envelope of service disturbance and we shall have to operate on the experimentally determined behavior of the system.

Preliminary tests have to some extent been confined to a 6J5GT tube. There seems to be some quantity that varies with the level of excitation and can be defined in terms of the number of cycles of excitation required to produce a given percentage of failures.

It is a matter of some interest perhaps that there is a 6J5GT tube and a 6J5WGT, the W standing for the ruggedized version. The ruggedized tube has been designed to withstand shock better than the nonruggedized one. Some of our results seem to show, in terms of steady-state vibration, that the nonruggedized version survives longer than the ruggedized tube.

In conclusion, it is hoped that, on the basis of the foregoing considerations, the acceptability of components can be determined in terms of their survivability under service shock and vibration. Then the correct design must be found to assure the survivability of the component and, therefore, of the equipment.

DISCUSSION FROM THE FLOOR

J. P. Walsh, NRL: I have a question to address to Mr. Crede. In his discussion, he mentioned briefly that he hoped to establish a relationship essentially between the shock spectrum and the damage. I wonder if he would be kind enough to give us just a few more words on this because his program is related to our subject for discussion at this session: the numbers derived from the shock spectrum which are useful in statistical analyses or in anything else we are trying to do.

Crede: It has been pointed out that the reed gage is very probably the most satisfactory and most simple device for measuring shock in the sense that there are reeds of various natural frequencies. We measure the deflection of each reed; we know what strength—or what deflection at least—each structure of the prototype equipment of the same natural frequency has to be designed for. While the reed gage is convenient in the sense that you attach it to the point where the measurements are to be made, the answer comes out directly. You merely measure the deflection of the reed and you have the answer.

However, there are many measurements available. Practically all the useful ones are measurements of either displacement velocity or acceleration as a function of time. These can be converted into the equivalent reed-gage response by some analog technique, either mechanical or electrical. We are using the electrical analog because it is much simpler to manipulate the various parameters. I am using here as a criterion, the similarity of the reed-gage response under service conditions with the response under conditions of shock testing in the laboratory.

We propose analyzing a number of records obtained in service and a set obtained from a shock machine, to find the shock spectra. If the two sets turn out to be similar we would be more surprised than anyone, but perhaps the results will indicate what changes in the nature of the shock pulse can be made to make the shock machine spectra more nearly in accordance with the spectra from the service condition.

That, of course, introduces problems of repetition about which a great deal could be said.

J. Steinman, Hughes Aircraft: Mr. Mintz mentioned the fact that some of the tubes tested failed immediately on entering the vibration tests while others lasted for quite a good length of time.

I would like to point out that unless you know the entire history of the tube from the time it left the factory until the time you begin tests, you are liable to obtain results which are quite erroneous because the tubes themselves prior to the testing may have received quite an extensive amount of shock and vibration.

Mintz: I grant that it is a possibility. There are two ways to improve the situation: (a) better quality control from the tube manufacturers, and (b) determination of the extent of transportation shock, so that the tube manufacturers can protect against it.

J. Markowitz, Barry Corp.: I believe that at Sperry, tubes are burned in on a vibration table and vibrated for a period of time. The number of tubes that fail is a function of two things, quality control and damage in shipment. Both contribute greatly to the picture.

J. H. Hutcheson, Hughes Aircraft: We have carried on an extensive vibration program on vacuum tubes for missiles. We have found that on tubes which have been tested for compliance to JAN specifications for vibration, there is a tendency for the tubes to degenerate with long periods of vibration and shock.

Many of our missiles which have been through shock tests and extensive vibration tests will no longer qualify after excessive test time. In going back and repairing these tubes we find in checking them that they are no longer in compliance with the JAN specifications. We are trying to accumulate data at the present time to determine what we can do about this very complex problem.

* * *

MISSILE RELIABILITY IN JEOPARDY

R. B. Ashley, NBS, Corona

The protection of guided missiles from physical hazards likely to be encountered during handling and transit is discussed. A need for quantitative determination of such hazards to serve as a basis for design, test, and reliability evaluation is presented.

Weapon reliability is a prime objective in missile development. Reliability of a weapon system depends on the reliability of each of its components. Packaging and packing are essential for guided missile system components. It is not sufficient to develop a perfect bird if that perfection is not given adequate protection from the factory to the launcher. All too frequently the problems of packaging and packing are overlooked during the early stages of development and the reliability is seriously jeopardized as a result.

The national bill for packaging of all kinds runs around ten billion a year. Domestic shipping damages each year amount to over half a billion. Of this, one hundred million can be charged to shock and vibration during domestic rail transportation alone. The damage to military equipment shipped overseas is fantastic since the loss in military effectiveness cannot be measured in dollars. A weapon represents an investment of critical resources, manufacturing facilities, manpower, shipping space, and time. In transit some weapons also represent a hazard to personnel, property, and military security.

Protection must be provided against the environmental hazards of nature such as temperature, pressure, humidity, fungus, salt, sand, insects, and rodents. Protection must also be provided against damage due to physical or mechanical hazards such as shock and vibration during transportation and handling. With the exception of temperature, protection against the environmental hazards is best provided by using hermetically sealed containers. Rigid containers such as metal or laminated plastic give limited physical protection against crushing, puncture and bending. This leaves the hazards of temperature and the forces resulting from shock and vibration, which must also be brought to a tolerable level by the packaging and packing. Statistics on probable temperatures are adequate. Unfortunately the same cannot be said regarding shock and vibration.

Most packing and packaging is developed by cut and try methods and tested by rule of thumb based on experience with similar items. For any new item this is rather haphazard and very time consuming. The threat to the guided missile program should be obvious considering that a missile package may contain fragile instruments, delicate circuits, critical structural alignments, and possibly dangerous propellants and explosives. Each of these is sensitive to shock and vibration. How can we prevent damage when we do not know the exact nature of the damaging forces which may be encountered? Attempts to measure these forces have been none too successful except for specific applications. Package markings such as "Handle With Care" - "Delicate

Instrument" are meaningless to illiterate dock-hands and disgruntled draftees. Special equipment and crews are impractical. Overpackaging is expensive and wasteful of materials and space. Trial shipments are fine for final proof testing but are of little help on original design. Must we go through the long and costly trial-and-error method as usual or shall we use a scientific approach? It has been repeatedly recommended that the damaging forces encountered during transportation and handling be identified and evaluated. The investment represented by guided missiles calls for no less.

The steps in a proposed evaluation of transportation hazards are listed below:

a. Review and correlation of previous tests,

b. Development of instrumentation,

c. Fabrication of representative packages,

d. Shipment of instrumented packages,

e. Data reduction and correlation,

f. Promulgation of results.

The object of the evaluation will be to obtain a complete and comprehensive picture of the forces which must be overcome to prevent damage during transportation and handling. This will involve quantitative evaluation based on a statistical analysis of extensive measurement of these forces by means of instrumented packages shipped unattended on representative trips of up to thirty days. The forces to be measured will be the result of isolated shock, repetitive shock, sustained impulse, steady vibration, and pulsing vibration, individually or collectively in one or more planes. All forms of transportation should be included.

Proposed Instrumentation:

A recorder for measuring and recording the the required data might be made up of the following components:

a. Three pairs of barium titanate accelerometers or other sensing devices to measure accelerations or forces in six dimensions, shock up to 50 g's resultant, and vibration from 2 to 1000 cps.

b. A 1-kc source of reference time impulses.

c. A seven-channel tape recorder modified to record intermittently at about 15 inches per second. Tape length at least 4800 ft to run about 1 foot at a time.

d. A thyratron trigger circuit to actuate the recording tape when accelerometer output exceeds a preselected level equivalent to about 2 g's.

e. A memory or temporary storage to hold the output from the accelerometers until the tape has reached recording speed. This might be a continuously rotating drum on which the impulses are recorded, read off, and erased during each revolution.

f. Amplifiers as required.

g. A source of power. If the tape runs 1/10 of 1 percent of the time, power requirements are estimated to average about 25 to 30 watts. If possible, the power supply should be sufficient for 30 days. Spring drives may be used to reduce battery drain.

The packages to be instrumented should be similar to the larger missile shipping containers. These are large enough to practically eliminate space and weight limitations on the recorder. It is essential that the recorder be protected and isolated from direct influence of the forces being measured. Ballast may be required to give proper dynamic reactions. The package may include other types of recorders for comparative purposes or for supplementary information.

Two methods of analyzing the records are suggested. Segments of individual records can be studied visually by means of the FM Magnetic-Tape Transient Recorder (Reference 1). This device reproduces a stationary picture of the recording on an oscilloscope by means of a continuous tape and synchronous repetitions. The mass of recordings can be analyzed and reduced to statistics by analog computer. The accumulative results should lead ultimately to standard composite spectra of the magnitude, duration, direction, frequency, and phase relationships of the forces encountered during transportation and handling, with an indication of the probability of occurrence on any particular trip.

Quantitative information is essential to the comprehensive design, testing, and evaluation of the packaging and packing that is needed for economical protection. A method of obtaining

this information has been suggested. Active support by all agencies concerned with shipping damage will expedite the program. Until complete information is available in useable form, the reliability of guided missiles is truly in jeopardy.

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1. Shaper, H. B., FM Magnetic-Tape Transient Recorder, Proceedings of the IRE, Vol. 33, No. 11, p. 753, November 1945

DISCUSSION

Dr. M. G. Scherberg, WADC: It appears to me that you have dismissed too lightly the human factor. Labeling packages "Handle with Care" may create a hazard if the handlers suspect that the package has been protected. Curiosity would vanish very quickly, however, if one word—say, "Explosives"—were put in front of "Handle with Care." I think that this is partly a problem for the psychologists.

Ashley: I agree. During the past war, it appeared that the packages marked "Handle with Care" got the roughest treatment.

We might find it in the interest of the protection of guided missiles to combine the missile components into packages too heavy to be lifted.

J. Steinman, Hughes Aircraft: Since you are interested in obtaining the spectrum of the carrier, why is it necessary to use a particular missile? Each missile probably will have a particular type of springing element peculiar to that missile. The data you get from a particular run will be applicable only to one type of missile.

Ashley: I mentioned packages for different types of missiles only because of the variation in size. In considering missile packages from small boxes to those too large for a flatcar, we get the complete scope of package sizes.

I do not think that actual missiles should be used. The instrumentation is substituted for the missile and the forces are the excitation forces on the package, not on the missile.

Percy Ott, NOTS: Most of the available reports on field studies contain the idea that the vibrations obtained are hash. None of the reports I have seen tell you how many times, how often, and how hard equipments have been shocked or vibrated. The only work that approaches giving a real picture of the number of shocks at various levels of intensity is that started back in the twenties by the AAR, with simple peak accelerometers. The instrumentation seems crude to us; it does not provide data on frequency or amplitude. However, the reports of the AAR do give you a picture of how many shocks occur at various levels. The data fit extremely well under probability curves.

It has been pointed out at this Symposium that a ride recorder should give us information on shock and vibration in handling and transit. We have tried one of the four or five commercial ride recorders on the market at NOTS and have found that the commercial instrument does not suit our purposes. This device was intended to indicate the size of the shocks occurring in transit, and the time of occurrence, so that the reliability can be pinned down to one railroad, trucking company or handling group. The commercial ride recorder is not a scientific instrument; the actual number of g's experienced may be 100 or 0.01 times the recorded number. The instrumentation still must be worked out.

Ashley: It is encouraging to note that the preliminary estimate of the cost of the first device I outlined is in the neighborhood of $40,000. This is not prohibitive.

R. L. McKay, Army Engineers, Fort Belvoir: I would like to say that my distinguished colleague, Mr. Frank Smallman, of the Vitro Corporation in Silver Spring, has spent two years studying this particular problem. In my opinion, his work adequately covers the situation.

J. Markowitz, Barry Corp.: There are some points that I would like to make in regard to the business of getting material where it is supposed to be used, in a useable condition. I have learned from the Signal Corps that the quickest way to unload a case from a truck at a point under fire is to rope a tree, put the rope around the case, and drive the truck away. This shows that the weight of a package does not eliminate the possibility of severe jolts.

The Signal Corps is making an extensive study of shock-mounted transit cases. They want to know what forces actually enter the case. For example, a certain amount of crushability on the corner of a case is definitely desirable; under a blow, the dissipation of straining energy is a great saving factor in reducing the damaging effect to equipment inside the case.

If an analysis does not include data on the effects of the case and of the bracket attachments, the picture is not complete.

Ashley: I appreciate that point of view. However, I feel in the laboratory or through calculations, we can determine the effect on the contents of forces on the shipping container itself. First, we must know the nature of the hazards and the probability of their occurrence.

I might make one other point in reference to the recorders. We get accelerations in g's but we obtain no information about the duration of the shock. One thousand g's operating for one ms are not going to do as much damage as 10 g's operating for 10 ms.

G. S. Mustin, BuAer: I would like to support Mr. Ashley's thesis that we need more data on the actual conditions prevailing in shipment.

In addition, I would like to bring to your attention the fact that in a recent survey for the RDB, the ground handling gear and the check-out gear looked like worse problems from the package design standpoint than the missile itself. In the packaging of guided missiles the entire weapon system must be considered, not just the bird itself.

* * *

SOME FACTORS IN THE DESIGN OF PACKAGE CUSHIONING

R. B. Orensteen, 1st Lt. USAF, WADC

Two basic problems in the design of package cushioning are discussed. First, given a particular material, what should be the thickness and area of cushioning? Second, what cushioning material should one use among the several available? The technique of using ratio-of-stress-to-energy curves in solving these problems is demonstrated. An example is given showing specific application to glass fiber cushioning for selecting optimum density, thickness, and area.

INTRODUCTION

In the shipment and storage of materiel, protection against shocks produced by rough handling and transportation is of primary concern to the packaging engineer. Cushioning is a means by which protection is provided.

In order to design economical package cushioning which will afford adequate protection of an item against shock, three main factors must be considered:

1. Hazards likely to be encountered in shipment and handling,

2. Fragility of the item,

3. Protection afforded by alternative methods of package cushioning.

The first factor, hazards in shipment and handling, is often simulated by dropping packs from various heights on a hard surface. The height of drop is usually chosen to represent the worst and/or most frequent hazard expected.

Very little reliable information seems to be available about the fragility of items to be packed. Nevertheless, some index of item fragility is necessary in a rational design of package cushioning. The index most often used is the "g factor." The g factor is the maximum allowable deceleration of the item before damage occurs. The larger the g factor, the less fragile an item. Large forces produce high deceleration, and in general, the larger the forces an item can withstand, the less fragile it is considered to be. The fragility of an item is often spoken of as so many g's, a multiple of the acceleration due to gravity.

Although more may be known about package cushioning than about either item-fragility or hazards in shipment and handling, there is still much to be desired in the proper use of data on cushioning materials. The purpose of this paper is to demonstrate a technique by which static cushioning data may be used in designing economical packs, consistent with the requirements of protection to the item. The static and dynamic behavior of many cushioning materials, for example, the cellulosic, is sufficiently similar to justify using static data in cushion design. Often, static data provides a margin of safety in design. Where static data is completely inadequate, as might be the case with many rubber materials, it is thought that dynamic data could be used in a similar way.

SELECTION OF CUSHION THICKNESS

Data which are useful in the design of package cushioning can be derived from the stress-strain

curve for the material. The stress-strain curve indicates how the displacement of the material changes when the load is varied. Figure 1 shows stress-strain curves for various densities of glass fiber cushioning. The area under the stress-strain curve is the energy stored by the material when it is compressed. For each stress value the material will store a particular amount of energy. The ratio of stress to energy is very useful in package cushion design. Figure 2 shows how the ratio, stress/energy per unit volume of cushioning, varies with stress for a 5 lb/cu ft glass fiber material.

The importance of the ratio, stress/energy per unit volume, is apparent when one examines the formula from which thickness is determined.

$$t = \frac{hf}{ge_f} \tag{1}$$

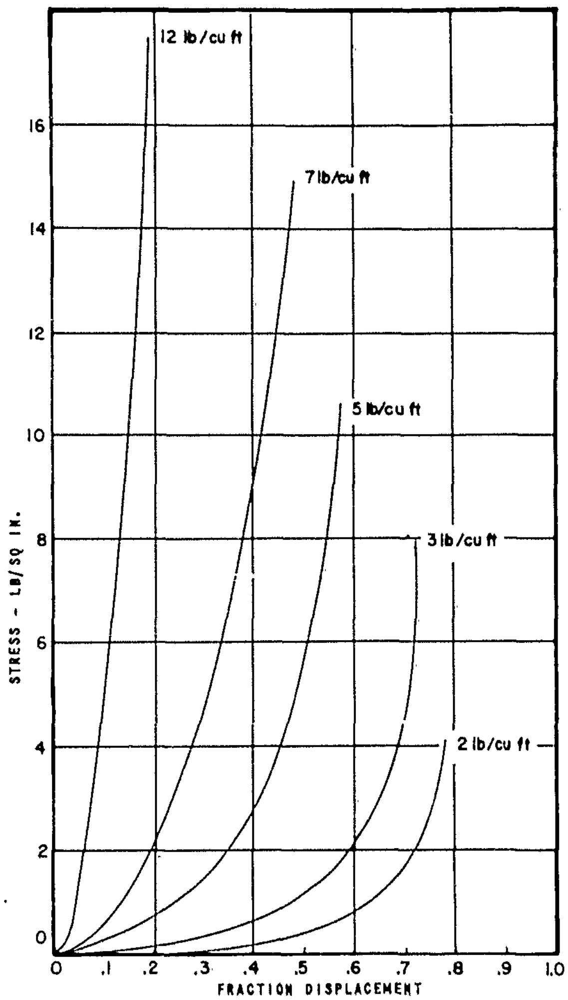

Figure 1 - Stress vs. strain for various densities of glass fiber cushioning

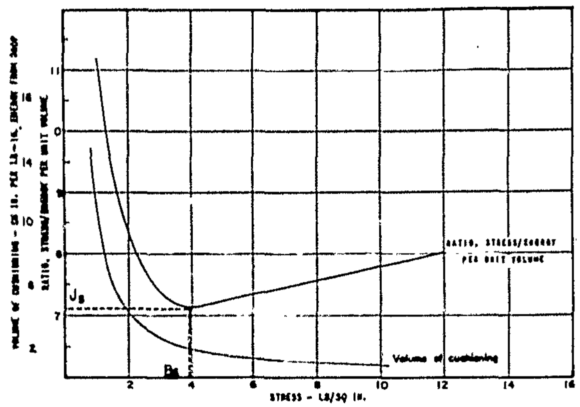

Figure 2 - Ratio of stress to energy per unit volume vs. stress for 5 lb/cu ft glass fiber cushioning

where t = thickness,
h = height of drop,
g = fragility factor in g's,
e_f = energy per unit volume of cushioning,
f = maximum allowable stress acting on the cushion.

The maximum allowable stress $\underline{f}$ is approximately equal to the static stress due to the weight of the item, multiplied by the fragility factor:

$$f = \frac{WG}{A}, \tag{2}$$

where W = weight of item,
A = area of cushion in contact with the item.

The derivation of the formula can be found in Wright Air Development Center Technical Report 53-43. The thickness formula is an approximation based on a fragility factor very much greater than one, and a cushion displacement small in relation to drop height.

In order to determine the proper thickness of a given cushioning material, it is necessary to have the following information:

1. Height of drop against which the item is to be protected,
2. Fragility of the item expressed in g's,
3. Ratio-of-stress-to-energy curve for the material.

Example: Assume an item with the following characteristics is to be cushioned against a 30-inch drop with a 5 lb/cu ft glass fiber material:

weight = 25 lb
area of cushion = area of item side = 75 sq in.
fragility factor = 30 g's.

The maximum allowable stress as determined by using Equation (2) is 10 lb/sq in. The ratio value from Figure 2 which corresponds to a stress of 10 lb/sq in. is 7.8. Substituting in Equation (1), the thickness is determined.

$$t = \frac{30 \text{ in.}}{30} \times 7.8 = 7.8 \text{ in.} \qquad (3)$$

SELECTION OF CUSHION AREA

The selection of the optimum design for package cushioning requires consideration of cushion area as well as cushion thickness. The factor of cushion area is one over which the designer has a wide latitude of choice. By means of interior cartons, die-cut pads, etc., he may vary the area of contact between the item to be packaged and the cushioning material. It is the weight of the item distributed over this contact area which in part determines the design of the package cushion. It is necessary, therefore, to determine criteria of cushion area selection in order to minimize cushioning volume and container cubage for economical package design.

If one examines Equation (1) from which thickness is determined, it is evident that the thickness $\underline{t}$ is a minimum when the ratio f/e_f is a minimum. The value of the minimum ratio could be used to determine the least possible thickness of cushioning when the fragility factor and height of drop are given. The minimum ratio could be substituted in Equation (1) and the least thickness could be determined. By substituting the value of the stress $\underline{f}$ corresponding to the minimum ratio into Equation (2) and solving for A, the cushion area is determined. The minimum ratio value is called J_S and the corresponding stress, B_S.

Are J_S and B_S proper criteria of cushion design? Using them will result in minimum thickness of cushioning. But will minimum thickness always provide minimum cushioning volume and minimum container cubage?

Cushioning volume varies inversely with energy per unit volume of cushioning material. Thus cushioning volume also varies inversely with stress because energy increases when stress increases. A curve showing how cushioning volume changes with stress is added to Figure 2. Inasmuch as the thickness required varies directly with the ratio f/e_f, the ratio curve can also be considered as showing how thickness varies with stress.

Figure 2 shows that the least value of the ratio J_S and the minimum cushioning volume are not necessarily consistent objectives. In terms of minimizing cushioning volume it would be advisable to increase stress as much as possible. In terms of thickness, however, there is an optimum stress. The use of J_S and B_S for optimum cushion design has been primarily a result of work done by R. R. Janssen of North American Aviation, Inc.

To design cushioning for minimum cushioning volume means designing for maximum practicable stress, and thus the maximum practicable energy absorption per unit volume of cushioning material. This can be achieved by reducing the area of contact between the item and the cushion. Would it be advisable to reduce the area of cushion to less than the area of the item itself? Yes, but perhaps only when both volume and thickness are being reduced. When the area of the item is greater than the cushion beneath it, container cubage is determined by the cushion thickness and the item area rather than by the cushion area. To increase cushion thickness by reducing the area of contact would reduce cushion volume at the expense of increasing container cubage. This condition would exist if the stress is increased beyond B_S.

It can be concluded that if the stress $\underline{f}$ is less than B_S, it would be advisable to increase the stress to B_S because thickness, cushion volume, and container cubage are reduced. The stress can be increased beyond B_S, but should be undertaken only when it is determined that the additional container cubage resulting from increased cushion thickness is compensated costwise by the reduction in cushioning volume.

Remaining to be discussed is the case where the maximum allowable stress $\underline{f}$ exceeds B_S. Should we reduce the stress to B_S? No, it would not be advisable to do so. Not only would cushion volume increase, but container cubage as well. However, if the stress $\underline{f}$ compresses the cushion to a point "near" bottoming, it might be profitable to reduce $\underline{f}$ even at the expense of increased container cubage. A better alternative might be to select a stiffer cushioning material.

In summary, there are occasions when it would be advisable to change the stress $\underline{f}$ of an item on

its cushion. However, there are also cases where the stress should not be changed. Table 1 summarizes recommended procedures in cushion area selection. When the number of g's that an item will withstand is given, the only way of changing the stress is by changing the area of contact between the item and the cushion. Substituting the desired stress in Equation (2) will give the area of contact. This area of contact, as well as cushion thickness, should be chosen consistent with the objective of economical package design.

TABLE 1
Recommended Procedures in
Cushion Area Selection

To Minimize Cushioning Volume and
Container Cubage

$f < B_s$	Reduce cushioning area so that stress equals B_s. Increase $\underline{f}$ beyond this point only if the additional container cubage is compensated costwise by the reduction in cushion volume.
$f > B_s$	Do not change cushion area unless stress $\underline{f}$ is "near" bottoming. If near bottoming either reduce $\underline{f}$ or select another material.
$f = B_s$	Do not reduce $\underline{f}$ by increasing cushion area. Increase $\underline{f}$ beyond B_s only if the additional container cubage is compensated costwise by the reduction in cushion volume.

SELECTION OF CUSHION DENSITY

One of the most interesting aspects of using glass fiber cushioning material in packaging is the wide range of properties that can be achieved. Among other means, cushioning properties can be altered by changing the glass fiber diameter, bonding material, amount of prestressing, and density. This section will deal with density of material as a factor in package cushion design.

Although this paper is a discussion of only a particular material, glass fiber cushioning, it still can be regarded as dealing with the general problem of the choice of cushioning material from among those available. Within the range of densities available, glass fiber cushioning conceivably provides an infinite number of cushioning properties, although practically speaking this number is of course limited. In the discussion that follows, density will be the parameter used to vary glass fiber cushioning properties. However, the choice of parameter here is not as important as the technique of relating a parameter to the ratio of stress to energy.

Figure 1 illustrates the variation in stress-strain curves which can be obtained with different densities of glass fiber material. A ratio-of-stress-to-energy curve can be derived from each of these stress-strain curves. A family of ratio curves for glass fiber cushioning is shown in Figure 3.

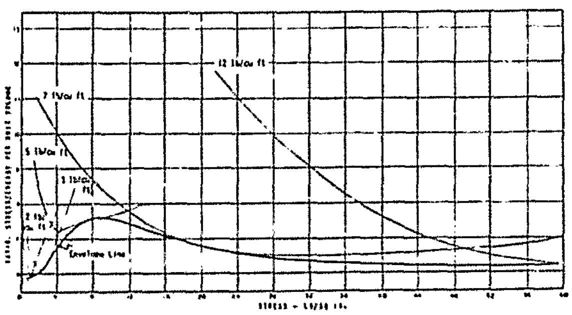

Figure 3 - Ratio of stress to energy per unit volume vs. stress for various densities of glass fiber cushioning

The points of most interest on the curves of Figure 3 are the least possible ratio values for any given value of stress. For a stress of 6 lb/sq in., the least possible ratio value, for the individual curves shown, is about 7.4 and is on the 5 lb/cu ft density curve. A point on either the 3 lb/cu ft or 7 lb/cu ft curve for this stress value would give a higher ratio of stress to energy. It has been pointed out that thickness varies with the ratio of stress to energy. When one can choose the density, higher ratio values than 7.4 for the given stress are of only incidental interest. If the choice of density is not open, the problem is entirely different and should be treated as in the previous discussion on cushion area and thickness selection. Notice that the least ratio value just determined for a 6 lb/sq in. stress is not the minimum ratio point of the density curve on which it lies. Nevertheless, 7.4 is the least possible ratio value among all of those plotted for the given stress.

Whereas the minimum points of the individual curves are of much concern when a particular density of material is considered, they are only of secondary importance when density can be varied. A line passing through these points is clearly not the answer; there will always be a ratio value on a lower- or higher-density curve more desirable than any minimum point one might choose. Of greater value is the line of

least possible ratio values. This is the envelope, the line tangent to the family of curves. The envelope represents the limit of the least possible ratio values as the number of curves approaches infinity. Any ratio-of-stress-to-energy curve corresponding to a given density would be tangent to this line.

The envelope is very useful in the design of glass fiber package cushioning. Given the stress, one can choose the least of all possible ratio-of-stress-to-energy values from the envelope line. The thickness can be calculated by substituting the ratio value in Equation (1). The individual density curve tangent to the envelope line at that particular stress would give the required density.

Rather than complicating Figure 3 by adding more individual density curves, a different approach will be taken. Two design curves, Figures 4 and 5, will be used in place of the envelope line. The relationship between density and stress is more clearly shown in Figure 4. The stress values corresponding to the tangent points have been plotted against the density values of the curves tangent at each point. A smooth curve has been drawn through these points. Hence, it will be possible to estimate the density required for points on the envelope line where no individual density curve is shown in Figure 3. A curve showing least possible ratio values from the envelope line plotted against density is shown in Figure 5.

The design procedure using Figures 4 and 5 is simple. From Figure 4 one can select the density of material which, among all others, will require the least thickness of cushioning for the given stress (the cushion area being assumed as

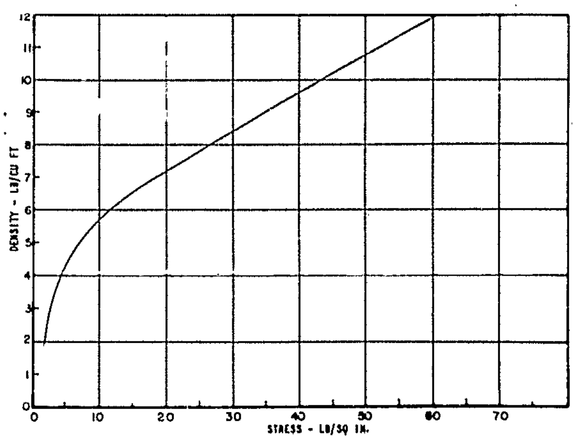

Figure 4 - Density vs. stress for glass fiber cushioning

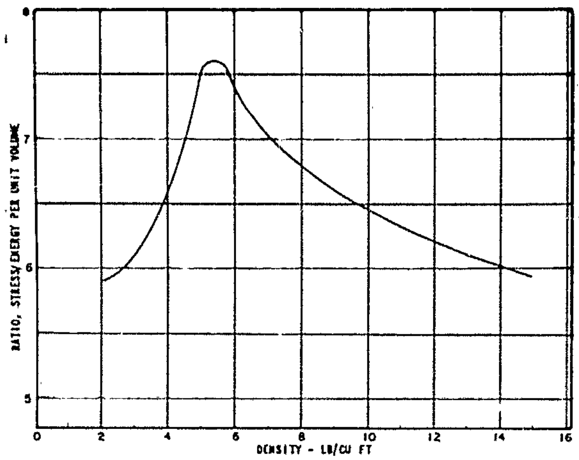

Figure 5 - Ratio of stress to energy per unit volume, vs. density for glass fiber cushioning

that of the item side itself). Once the density is chosen, the thickness is easily determined from Figure 5. The least ratio f/e_f is read from the curve and substituted in Equation (1) to determine thickness.

Example:

Assume an item with a fragility factor of 20 g's, a weight of 50 lb, and an area of side 50 sq in. The stress from Equation (2) is

$$f = \frac{50 \text{ lb} \times 20}{50 \text{ sq in.}} = 20 \text{ lb/sq in.}$$

From Figure 4, the density corresponding to this stress is about 7 lb/cu ft. From Figure 5, the ratio f/e_f for this density is about 7. The thickness of cushioning, substituting in Equation (1) and assuming a 30-in. drop,

$$t = \frac{30 \text{ in.} \times 7}{20} = 10.5 \text{ in.}$$

Warning - The above procedure should not be reversed. Given a density of material, a ratio value can be determined from Figure 5. But thickness of cushioning can be calculated from this ratio value only in the event that the stress happens to correspond to this point, or is altered by an area change to do so. The correct procedure would be to determine the ratio value from the individual curve for that density of material. Then the procedure with regard to area selection can be followed as previously discussed.

OPTIMUM CUSHION DESIGN

In the previous discussion of cushion density it was assumed that only the density and thickness are selected. The cushion area was given as equal to the flat area of the item itself on the side in question. It is interesting to consider the possibility of cushion area changes simultaneously with density and thickness selection.

The envelope line of Figure 3 shows the relationship between stress and ratio of stress to energy per unit volume when designing for minimum thickness. As the stress and density increase, the ratio values increase to a point and then decrease. Since the fragility factor is given, the only method of changing the stress is by changing the area of contact between the item and cushion. To increase stress, reduce area; to reduce stress, increase area. By changing the cushion area one can obtain various ratio values.

A ratio value in the low stress range can be duplicated in the high stress range with a high density material. For example, a ratio value of about 6.7 is obtained with a stress of 4 lb/sq in. and a density of 4 lb/cu ft. This same ratio could be obtained with 8.5 lb/cu ft material and a stress of 30 lb/sq in. What would be the advantage in increasing the stress, reducing the area, and increasing the density of material?

Assume a case where we obtain the same ratio of stress to energy per unit volume by means of a low stress, low density material and a high stress, high density material. How have container cubage, cushioning volume, and cushioning weight been affected? When the cushion area is less than that of the item side itself, container cubage will generally be reduced only when cushion thickness is reduced. There is no benefit in this respect—thickness will remain unchanged since the same ratio is assumed for each of the stresses.

Figure 6 shows what happens to cushion volume and cushion weight as one moves to higher stresses and higher densities. As cushion area is reduced, thereby increasing stress, cushion volume is reduced. Remembering that to achieve minimum thickness, higher stresses require correspondingly higher densities, the weight of cushioning material may either increase or decrease depending on the volume reduction as compared to density increase. Multiplying the volume for a particular stress by the recommended density for that stress, weight values

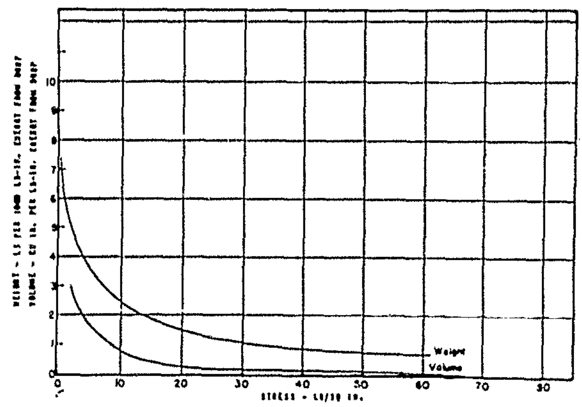

Figure 6 - Weight and volume of cushioning vs. stress

are obtained. These values have been plotted in Figure 6.

Reference to Figure 6 indicates that both cushion weight and volume are reduced as one reduces cushion area to increase stress. From a cost standpoint, cushion weight is an important factor. Material prices are in some way related to the amount of raw material that goes into a given volume. Thus the weight curve indicates a downward trend of material cost as stress is increased. Moreover, insofar as shipping costs are related to weight, these costs would be reduced.

The envelope line of Figure 3 shows another interesting fact. By increasing density and stress beyond a certain point, one could obtain a reduced ratio of stress to energy per unit volume. This means not only reduced cushion volume and weight, but reduced cushion thickness and container cubage. In a practical sense, this means that perhaps the most economical way to use glass fiber cushioning (and possibly other materials) is to use very high density material of small cushion area. Small strips, cylinders, and side and corner pads might be utilized for this purpose. There would be limitations, however, in how far this procedure could be followed. For example, a very thick cushion of small area buckles easily. However, there may be ingenious ways of circumventing such problems.

Examples (same data given as in previous example)

In using a 12 lb/cu ft material instead of a 7 lb/cu ft material, a change in the cushion area

and thickness is required. A 12 lb/cu ft material would require a stress of 60 lb/sq in. (Figure 4) and a ratio f/e_f of 6.2 (Figure 5). The cushion area would be calculated from Equation (2) as follows:

$$A = \frac{50 \text{ lb} \times 20}{60 \text{ lb/sq in.}} = 16.7 \text{ sq in.}$$

The cushion thickness would be calculated from (1) as follows:

$$t = \frac{30 \text{ in.} \times 6.2}{20} = 9.3 \text{ in.}$$

From the previous example, the requirements for a 7 lb/cu ft material are:

A = 50 sq in.
t = 10.5 in.

Thus, without reducing the protection to the item, a cushion of dimensions 50 sq in. x 10.5 in. has been replaced by a cushion 16.7 sq in. x 9.3 in. Cushion volume has been greatly reduced. Container cubage has been reduced because the cushion thickness is less. But more significant in the given example is the reduction in cushion weight. Multiplying the volume of cushioning required by the density gives the weights shown in the following table:

Density	Volume	Weight of Cushioning
7 lb/cu ft	0.30 cu ft	2.1 lb
12 lb/cu ft	0.09 cu ft	1.1 lb
volume reduction	0.21 cu ft	
weight reduction		1.0 lb
percent volume reduction	70%	
percent weight reduction		48%

A saving in cushioning volume (not necessarily container cubage) of 70 percent and a saving in cushioning weight of 48 percent resulted from a reduction in cushion area. This is in addition to reduction in container cubage resulting from reduced cushion thickness.

SUMMARY

In the foregoing discussion, a technique has been demonstrated by which static data can be used in the design of package cushioning. It was shown that the selection of cushion thickness, density, and area can be determined by use of the design curves of density vs. stress and ratio of stress to energy per unit volume vs. density. This procedure can easily be extended to the general problem of material selection. The designer would be provided with design curves for all package cushioning material. From these he could easily determine the design requirements for each material. The selection of the material now becomes a problem of economics. Conceivably one might have to choose between a material of heavy weight and little volume and material of light weight and large volume. The technique herein discussed should provide assistance in making an economical package cushioning decision.

It is hoped the data of this paper will be useful in design of package cushioning; but this report is more to illustrate a technique than to provide design data. To anyone who might use the data of this paper, a word of caution. The procedures outlined will give only an approximation to the correct design. This approximation should be verified, when practicable, by test or service experience. This is some improvement over the prevalent trial and error method of package cushion design. The long run objective is the possibility of predetermining package cushion design, without subsequent testing, to provide the necessary protection for the item.

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DISCUSSION

R. L. McKay, Army Engineers, Fort Belvoir: In our laboratory, we feel that there is a distinct advantage in using nine accelerometers positioned always in the same manner around the contained item in every test that we make.

It appears that Lt. Orensteen is making the assumption that the cushioning material remains in contact with the contained item at all times. We have thousands of curves showing conditions when this assumption does not hold true. For example, a particular guided missile whose axis of strength lies along the axis of fire can take 40 g's along its main axis. However, the missile tends to crumple like an eggshell along any axis at right angles to the axis of fire.

It would be extremely difficult, if not impossible, to use the type of cushioning which you have described so that no matter what happens to the container the end effects of the shock and vibration seen by the center of mass of the contained item is seen only along the strongest axis. We feel that we can do this through a mounting system—a rubber shear type or sand or something of that nature—that has a well-known relationship to the center of mass of the contained item.

I might add that we have used the methods of electrical engineering in our analysis as well as the theories of transients for several mechanisms. We feel we are making slow but sure progress in tying in the packaging problems with another field which is well-developed and well-grounded in mathematics.

Orensteen: I can understand that, for example, in a package where the cushioning is somehow precompressed during the drop, the item might leave the cushioning above it. At some point during the drop one has to make calculations as you have done for each side or axis of the item. If you have a certain axis of strength, the reason for putting cushioning, or whatever means we are providing, perpendicular to that axis, is that less protection is required from the forces taken for directions other than those perpendicular to the axis. Perhaps one would have to introduce a different g factor or some different index of fragility depending on the calculations one is making.

Also, there are, certainly, many situations where cushioning is not the best and most economical method of doing the job. Unfortunately, in the military many of these other systems—spring systems, shoe pads, and so forth—have their disadvantages from the point of view of people in our depots who have to work with these systems. It is simpler to have a packer take a radar instrument and lay it into a premolded packaged cushioning than it is to rely on his judgment to shock mount it properly, and attach the item to a bracket.

McKay: I will go along with your basic concept. However, I think you will find that you need a mounting system whose characteristics are known for all conditions.

There are certain conditions of resonance which you can readily define in the transient phase of the problem. However, when you consider steady state, it is very difficult to analyze the situation. There will be certain resonant modes.

Certain resonant conditions are beyond control when you are using cushioning. These may be controlled more readily if you have a mounting system which has constant characteristics for both the steady state and the transient introduction of energy. It seems possible to design a permanently connected mounting system that has such characteristics.

Orensteen: I have presented only a part of the problem, an attempt to protect the item against a few shock transients of a certain type. I have not dealt with the problem of the critical frequencies that may occur in a cushioned package. Some might very well be harmful to the item itself and if there are such frequencies I feel that much can be done to improve conditions if package

cushioning is being used. As you suggest, other systems might be necessary.

K. Kuoppa-Maki, NBS, Corona: You mentioned precompression. This analysis of yours would be still more valuable if one could follow the change in compression during transport. If you take the same cushioning system and make a run analysis of a trip across country, you can get a more realistic picture.

Orensteen: With cushioning material you do have the problem of set. That is, after the package and item have vibrated or have been dropped several times in the package, the cushioning material may take a set such that it has new properties. One problem not discussed in my paper is the determination of the amount of precompression necessary for such material so that after several successive drops, the material will still provide protection to the item. Many materials take considerable set; with other materials set is of secondary importance.

G. S. Mustin, BuAer: It seems to me there is the fundamental assumption in this paper and in all the other papers on packaged cushioning that I have read, that the drop is flatwise. It appears to be inherent in the calculation analysis by Mindlin. Unfortunately, the rotational drop is a far more common type of drop, especially when the container gets to be fairly large.

I have found it difficult to handle the rotational drop by using the Mindlin method, and was forced to go back to the load-deflection curve and the load-energy curve to achieve any useful design. Unfortunately, when you have a rotational component in a diffused cushion, there is as yet no analytical method of handling it, although the problem is being studied at the Naval Research Laboratory. I prefer to regard the Mindlin method as optimum, as it was originally defined by Janssen, and to use it as a guide in selecting a cushion which will give the optimum characteristics; but in computing the accelerations on an item you cannot use it, as far as I know.

Orensteen: I have no quarrel with anyone who wants to refine the method of finding, for example, what happens when a rectangular package is dropped on the corner. However, what benefit is derived from calculating the accelerations along some axis if we don't know anything about the item itself, what protection the item has, or what fragility it has along the axis?

Secondly, we have a safety factor in many of the designs, introduced by the flexibility of the container itself. During impact the container may crush and absorb energy from the drop. The value of these considerations is determined by the particular refinement that you want to make. You can refine my work by making less assumptions; or you can go the other way and make more assumptions depending upon the situation and the answer you are after.

I have never tried to use this analysis to find out what happens on the cornerwise drop. I have made some preliminary calculations which show that a design for the side of an item give considerably more than enough protection for the drop because the average thickness available on a corner and other areas is such as to provide sufficient protection.

Mustin: I think you are correct in stating that all the techniques give you a margin of safety wide enough so that you can handle these problems conveniently in your own day-to-day design work. However, to me there is the problem of finding the amount of clearance necessary to allow for the rotation of the item after impact, rather than going into a detailed discussion of the effects on the item itself.

* * *

CONCEPTS AND NEEDS IN THE OVER-ALL PROBLEM FOR SHIPPING MISSILES

H. C. Bolon, Redstone Arsenal

This discussion presents concepts and needs relative to the over-all shipping problem concerning guided missiles and missile-rockets. The word "shipping" should be interpreted as meaning both shipping and storage. It seems highly improbable that any missile can be either shipped or stored without being packaged. Also, it appears uneconomical to design a container for each condition. Parameters established are very general and present only the basic guide lines for assuring safe transit and storage of missiles. Effective design requires a through knowledge of the item and of how it will be handled and used logistically.

Questions asked relative to the item are: (1) What are its structural characteristics? Is the basic structure designed to maximum or minimum strength of the materials employed? (2) How does the basic structure affect the components placed within it? Are the components rugged or delicate? Does the structure supplement the component or does it place additional requirements on it? What and where are the allowable loadings under impact? What amount of steady-state vibration is the item capable of withstanding? (3) What is the size, weight, and shape of the item? (4) Is the item designed to withstand environmental conditions, i.e., heat, cold, humidity, dryness, wind-blown snow, wind-blown sand, deterioration caused by salt air and industrial atmospheres?

Then it is necessary to ascertain how the item would be handled through its logistics. Questions asked relative to this phase are: (1) Is the item designed for use in zone of interior, overseas, or both? (2) How will it be handled, shipped, and stored? What protective measures are necessary to supplement those present in design of the item? What degree and type of isolation is required to protect the item during handling, shipping, and storage? Does the item in its container lend itself to transport by truck, rail, ship, and aircraft in Phase II or III airborne operations? Does the containerized item lend itself to handling with currently available equipment or is special equipment required? How far forward can the item move and still remain protected against shock, vibration and weathering action? How do currently available storage facilities (defined as warehouses, sheds, igloos, magazines, and other areas for storage of dangerous materials) lend themselves to handling of the containerized item? Are overhead lifting and conveying equipments available? In the absence of overhead equipment is ceiling height compatible with economical storage? Do warehouses designed for less voluminous packages, lend themselves to storage of this material? Can aisleways be rearranged to permit operation of handling equipment? Will space be available for necessary inspections and maintenance-in-storage operations? Is it desirable to supplement the container with some type of handling fixture?

The designer now comes to the final questions. How well will the container fit the needs of the ultimate user? Does the container fit into the tactical handling and equipments picture? Does it allow use to the assembly area? Can it protect the item to the launching area? If it can, and if all operations required between the assembly line and the launching area, can be performed with a minimum of cost, time, and man hours expended, the job has been well done.

The foregoing discussion has established factors governing design. The question now arises as to when the design must be conceived. Obviously, this should be at or near the time

when the item is conceived. The above questions need to be answered. They affect the entire system and plan of logistical handling. Unless the item incorporates certain features it will not be possible to package it efficiently and economically for its intended use. Structures, components, and systems should be designed to be as rugged as possible compatible with weight. A minimum consideration in any design would be a sufficient number of attaching points to allow efficient incorporation of an isolation system. Preferably those points should be located in a main structural member or members.

ARMY ORDNANCE CRITERIA

Army Ordnance has developed criteria, based on detailed study over a period of time, for the design of guided missile shipping and storage containers. Indications are that these criteria are going to allow standardization of missile containers for our storage and supply system. They are very general but since they set forth certain standards that are required in missile containers, it is believed that they are worthy of review and of statements as to what may be accomplished when they are used.

SCOPE

These criteria outline the basic general design principles of shipping containers for guided missiles and major components.

CLASSIFICATION

For the purpose of design, containers are segregated into the following types and classes:

Type I - Containers for missiles employing electronic guidance systems

Class A--Containers for missiles up to 40 in. in diameter and/or 40 ft in length

Class B--Containers for missiles exceeding 40 in. in diameter and/or 40 ft in length

Type II - Containers for missile-rockets other than those employing electronic guidance systems.

DESIGN REQUIREMENTS

Design of containers shall meet the requirements as outlined in AR 740-15, paragraph 3, and SR 705-70-5. All containers shall be designed in such manner as to incorporate the degree of preservation and protection required to withstand the rigors of weathering actions and all types of climatic conditions encountered in global distribution. Provisions shall be made for protecting the item against the various hazards expected to be encountered when exposed to the handling and storage conditions generally encountered in military supply, maintenance, and distribution systems. Design shall meet the requirements imposed in handling and transit of all modes of transportation; motor truck, rail, ship, and air as well as the conditions encountered in storage (storage conditions are defined as warehouse, open shed, and open on field).

Type I - Class A Containers

1. Material

   The materials used shall be such as to produce a rigid, reuseable-type container (metal, its equivalent, or better).

2. Required features (not necessarily in sequence of importance).

   a. Must provide ease in opening and resealing, with a minimum of seal area in all cases (e.g., "end-opening cylindrical container"). Sufficient structural strength should be provided to prevent undue distortion and puncture or damage to the item itself during handling, transit and storage. The outer structure must be of such dimensions as to permit transportation by commonly used methods of transport and must be capable of Phase II or III airborne operations.

   b. Instrumentation, ports, and openings shall be provided to allow checking of the contents and interior of the containers (e.g., air fill and pressure relief valves; pressure gauge; humidity indicator; registering accelerometer; ports and openings for installing batteries or cables for circuit checks). Instrumentation shall be grouped in one panel in the end bulkhead. Other openings shall be limited to a maximum of two.

c. The isolation system shall be capable of providing damping properties required to protect the item within the container from both impact and steady-state vibration encountered during handling, transit, and storage. The isolation system should be so designed as to preclude its removal prior to the launching site area stage of missile transport.

At this point I believe we should look at some of the container configurations most commonly used that would meet the requirements of Type I Class A containers, either in part or in whole. Figure 1 illustrates three basic configurations being used or anticipated for use in the packaging of missiles at the present time—the casket type, the end-opening cylindrical type and the end-opening rectangular type.

The casket type usually splits at about the center line with relation to the vertical height of the container. Top and bottom sections are flanged in order to effect a seal. Overhead lifting equipment is needed with this container. In use with missiles the top half of the container generally must be removed and moved away from the area; the upper half of the missile is then accessible for maintenance or other work. When access to the lower half is required, it becomes necessary to remove the missile from the isolation system and place it on some kind of a work stand.

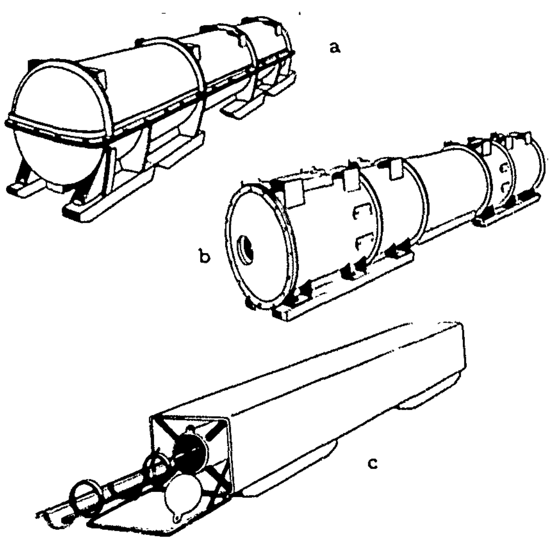


Figure 1 - Type I Class A Containers—basic configurations

a) Casket

b) End-opening cylindrical

c) End-opening rectangular

The end-opening cylindrical type using a track-in track-out arrangement allows withdrawal of the missile without breaking down stacked containers. Seal area compared to the casket type is relatively small. When removing the missile from the container a rack or stand is required but can be designed so that the missile remains in its isolation system. A container of this type lends itself to marrying with other parts of a missile (e.g., the incorporation in an assembly area of a Jato or Booster unit employed on a Nike or Terrier missile). This container can be used for the Corporal, Hermes, Talos, and Falcon missiles.

The third container shown in Figure 1 is the end-opening rectangular. Only one end opens on this container and again the missile is withdrawn from the container by a tracking arrangement similar to that for a file cabinet drawer. Here too the seal area is relatively small. Provisions must be made in the top and skids to insure tie-down and stacking. The LaCrosse is one of the missiles for which this container is intended.

Shock isolation systems in any of the above containers are selected on the basis of the item under consideration. They may be shear mounts, hydraulic mounts, mechanical springs, or any combination. The item and space required for the system are basic considerations. Assuming two systems will perform equally well, it is very important that the system requiring a container with the lesser cube be selected. During tactical conditions, space always becomes a prime logistic concern. This is particularly true when ship transport is required. It is also well to remember that shipping cost is computed on the basis of cubic volume. Figure 2 illustrates three applications of an end-opening container. The exterior isolation system (Figure 2a) will allow a slight decrease in total weight. It requires a supplemental handling rack in order to perform checks and maintenance operations. When supplemented with a covering, it will allow missile transport to the launching area in the isolation system. Figure 2b uses an interior isolation system. It too requires a supplemental handling rack for checks and maintenance operations. Protection to the item can be achieved to the assembly area and by using the handling rack, the missile can be sent forward to the launcher in the isolation system. The third container

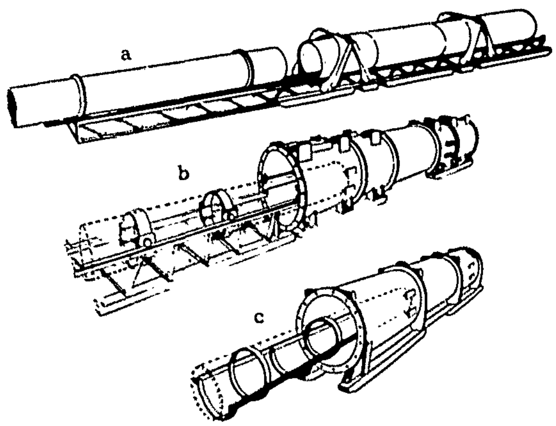


Figure 2 - Applications of an open-end container

a) Container with exterior isolation system

b) Container with interior isolation system

c) Container with interior isolation system (no handling rack required)

(Figure 2c) is one in which it is possible to roll the missile forward or 3/4 the missile length aft. In the aft position the missile remains on the track-in track-out rails. When it is necessary to remove the missile from the container it must be freed of the isolation system, and additional handling equipments are required. This container would allow transport to assembly area. Figure 3 shows an end view of the containers shown in Figure 2. Figure 3a shows hydraulic shock isolators, used in conjunction with linear springs to accomplish the over-all isolation system. Bolts through the container into the handling ring tie the missile into the isolation system during its period of storage within the container. Note in Figure 3b the positioning and tie-in points in which the missile in the isolation system is tied into the main structure of the container. Figure 3c shows the methods of track-in, track-out, and tie-in of the missile in the isolation system to the container.

Type I - Class B Container

1. Materials

The materials used shall be such as to produce the necessary protection compatible with and supplementary to the preservation applied to the missile itself (metal, its equivalent, or better).

2. Required features

a. The container must provide ease of assembly and disassembly and should be of such weight as to preclude the use of special handling or lifting devices. Sufficient structural strength should be provided to prevent undue distortion and puncture or damage to the item itself during handling, transit, and storage. The outer structure must be of such dimensions as to permit transportation by commonly-used methods of transport and must be capable of Phase II or III airborne operations.

b. Openings for access to the item within the container shall be confined to a maximum of two in addition to normal closures and the locations shall be computed on the basis of the enclosing structure itself. The locations of these openings shall in no way affect the structural strength of the container.

c. The isolation system shall be capable of providing damping properties required to protect the item from both impact and steady-state vibrations encountered during handling, transit, and storage. The isolation system should be so designed as to permit its use in the handling sequence up to and including if possible the launching site area.

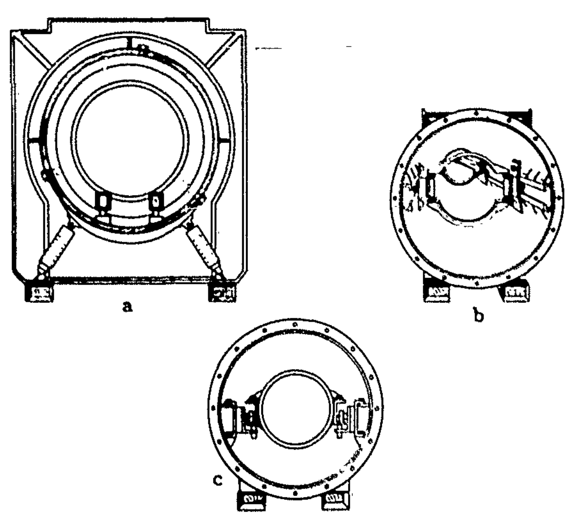


Figure 3 - End views of the containers shown in Figure 2.

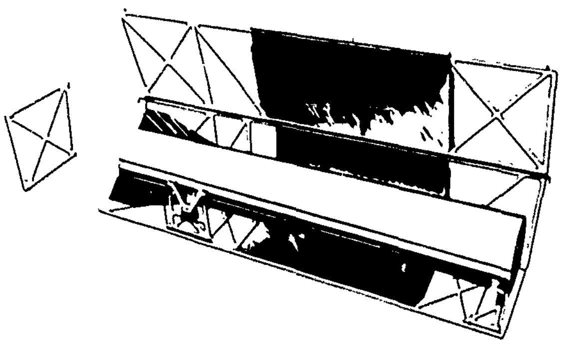


Figure 4 - Type I Class B Container

Figure 4 illustrates an application of this container. The diameter of the missile section is 5.85 ft (7.33 ft with fins attached) and the length is 38.75 ft. Fuel requirements for this intercontinental type missile dictate a minimum stress design in order to allow the required payload. This design requires a cradle to support the missile. The cradle must be an integral part of the isolation system. Due to the size and weight of the missile, it cannot be economically containerized completely as in Type I Class A application; therefore, its design must lend itself to a required level of preservation. Note that the function of this container is merely to supplement that preservation and affords only protection against physical damage. It also must provide a high degree of shock and vibration protection in the isolation system. The base section of this container performs the principal function; the sides, ends, and top are designed only to supplement that function. Missiles of this type and size present several interesting problems inasmuch as they are very close to allowable limits of transportation facilities. In flight, particularly in launch and boost stages, they are almost never subject to shock factors and the factors that exist are negligible. Steady-state vibrations are predominant considerations. Handling and transportation, however, introduce both impact and steady-state vibration. An example of a missile in this category is the Redstone.

Type II Containers:

Type II Containers are for missile-rockets (a coined phrase) not employing electronic guidance systems. A missile-rocket in this category is Honest John. The following features must be designed into all Type II containers.

1. Materials

   The materials shall conform to current Military Specifications MIL-P-6057, JAN-P-104, JAN-P-105, JAN-P-106 and their interrelated specifications. The containers shall be capable of providing protection required by the item; otherwise, containerizing shall be as indicated for Type I Containers.

2. Required features

   a. The container must incorporate features permitting ease of assembly and disassembly and should be of such weight and dimensions as to preclude the use of special handling or lifting devices. The item must be so positioned within the container as to prevent the likelihood of damage caused by puncture or structural failure during handling, storage, and transit. Its dimensions must be such that common methods of transportation may be utilized as well as Phase II or III airborne operations.

   b. Closures

   Final closure of the containers shall be effected in accordance with the governing specification except that in all cases the containers must be modified to include the use of screws, bolts, or lag screws for reuseability.

   c. Isolation system

   Generally, isolation will be accomplished by use of cushioning materials as specified in Military Specification MIL-P-6064. However, analysis of the item may permit the use of other commonly used materials of a non-hydroscopic nature such as creped cellulose wadding Federal Specification UU-C-843. Under no circumstance will it be permissible for a cushioning material to contact a metal surface when that material has a PH factor that does not fall within the range 6.0 to 8.0.

   d. Preservation of electrical and other critical components

   Such components shall be preserved and packaged within the container in accordance with Method II, Military Specification MIL-P-116. Under no circumstance shall barrier-material used in those applications be other

than Class I vapor-proof barrier-material Military Specification MIL-P-131. Efforts shall be made to hold the inclosed volume of the Method II package to an absolute minimum. Only under extraordinary circumstances will it be permissible to incorporate a whole missile airframe within this type barrier.

Figure 5 shows one method of accomplishing a Type-II Missile Package. The figure shows a method of anchoring, blocking and cushioning the item to the base of the container. The cushioning material is commercially available. The method of blocking and bracing, together with the cushioning, forms the isolation system. It is necessary in this application only to protect the missile section physically against damage during handling, storage, and transit.

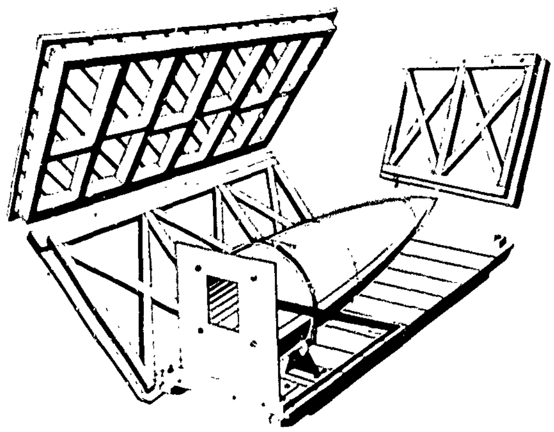


Figure 5 - Type II Container

TESTS

Tests shall be conducted on both Type-I and Type-II containers to verify the adequacy of the design for preserving and protecting the contents of the container. Guides to the selection of tests are found in Military Specifications MIL-P-116, MIL-P-6055, and MPD-113. Sufficient tests must be performed to assure not only the adequacy of the container design, but verification of that design to protect and supplement methods of preservation and packaging used in conjunction with the container and applied directly to the item.

PREPARATION FOR SHIPMENT

All containers must be capable of final closure in such manner as to insure their acceptance for safe delivery by truck, rail, ship, and air transport.

NOTES

It is believed that these general criteria, supplemented by detailed studies of the individual items, will give parameters for the design of containers for missiles and components.

Other features necessary for all containers not specifically covered elsewhere herein are:

1. The stated policy of the Office, Chief of Ordnance, is that, wherever possible, missiles will be packaged as a complete assembly. Disassembly should be held to an absolute minimum commensurate with the use of the item.

2. Wherever possible, all containers must be capable of being handled with commonly available equipment (e.g., fork lift trucks, cranes, slings).

3. Containers must carry sufficient markings (e.g., identification, caution, center of balance). Exterior surfaces of containers will be provided with adequate protection to withstand weathering action.

4. Features shall be incorporated in all containers to provide safe handling with fork lift trucks and lifting slings. Also, some method should be provided for tie-down and locking in stacking so that, where desirable, more than one container can be handled simultaneously.

5. All containers should be designed to provide in the most economical manner the maximum preservation and protection to the contents, with the lightest possible weight and minimum cubage.

6. Containers should be designed so that special tools are not required in opening, or closing, or checking in maintenance operations. All tools required for those operations should be readily available from normal issue channels.

Thus far we have presented concepts of design and some basic examples. There are many other possible variations. Missile packaging in its present state leaves much to be desired. There is a distinct feeling that unrealistic conditions are being imposed on the developer of the missile package. The criteria for package design are

usually expressed in so many g's in three planes and at certain stations of the missile. When you question these apparently ridiculous criteria you are answered that they result from studies of in-flight conditions. Checks of in-flight conditions show the figures given are in the order of seconds, while handling, storage and transit conditions are invariably in the order of milliseconds. More information is certainly desirable on this particular subject as costwise, materialwise, and timewise there are indications that we may be missing the exact criteria on some of our packages.

Another need to reduce the cost, material and time factors is some type of plastic material to be used in intimate contact with the missiles. This material should be capable of affording protection equal to Class I MIL-P-131 barrier material. It should be very light in weight, easily strippable, and should lend itself to application by brushing or spraying. It must be capable of withstanding prolonged periods of storage. Still another need is materials for shock isolation systems that will give better performance at the lower temperature range required in military packaging. Another material need is an inexpensive plastic that will lend itself to container constructions.

It is believed that additional information is needed as to exactly what happens to a missile during handling, storage, and transit. Further, it is indicated that the economical way to approach this problem would be on an inter-agency basis. This method of approach would allow several agencies to work on the over-all problem with each responsible for one particular phase of the investigation. In this way, time consumed for the total project could be greatly reduced.

The above discussion, I believe, presents a picture of missile packaging as of this date. True, it may vary from technical service to technical service, but, generally, there is a striking parallel throughout the military establishment.

DISCUSSION

G. B. Mustin, BuAer: Instead of asking a series of questions I would like to observe that the logistics of the business are going to determine the design of the missile container. This is a very important point; there is no standardization of logistics with various missiles, simply because the tactical requirement will vary depending on the anticipated need. Therefore, the word "standardization" should be taken with a great deal of salt. If you interpret the word to mean that all missile containers should look alike, that is going to be a frightful mistake. About the only statement to make about standardization in the missile field is to say that a shipping container for a very large missile must perform thus and such, or a large container for the general types of missile such as CORPORAL or REDSTONE must perform in such and such a fashion. If you get down to a very small container (we have been forced to consider containers as small as 2 in. x 2 in. x 2 in.), then it must perform in a given manner. The only standardization is going to be performance of certain sizes and geometrical configurations.

* * *

SIMULATING MILE-LONG RAPIDLY APPLIED ACCELERATIONS IN THE LABORATORY

J. H. Armstrong, NOL

The Rotary Accelerator, described in principle in Shock and Vibration Bulletin No. 14, is now in service at the Naval Ordnance Laboratory. A description of the machine as actually constructed is presented, with oscillograms of acceleration-time curves and results from its use in tests of rocket and missile component response to launching accelerations.

INTRODUCTION

A previous article on this subject[1] set forth the value of duplicating in the laboratory the effects of rapidly applied rectilinear accelerations of durations comparable to hundreds or thousands of feet of travel. Missile boost, rocket setback, and high-velocity water penetration are examples of such accelerations, and the response of weapon components to them is both difficult and expensive to determine by field weapon firings or rocket track-sled tests.

There are several laboratory methods now in use for simulating long-duration accelerations, all of which are based on the use of centripetal acceleration to provide an indefinitely long acceleration by motion in a closed path. Because of the proportionality of acceleration to radius, any machine used for such application must be large in relationship to the tested specimen if a reasonably constant acceleration is to be provided over the whole specimen. The various methods which incorporate each of the operational principles, together with their respective advantages and limitations, are given in Table 1. Each method has a broad field of usefulness in problems where its particular limitations do not invalidate the test results.

[1]Armstrong, J. H., "The Rotary Accelerator," Shock and Vibration Bulletin No. 14, p. 43, Dec. 1949.

TABLE I

Laboratory, Long-Duration, Rectilinear-Acceleration Simulators

Operating Principle	Principal Advantages	Principal Limitations
Conventional Centrifuge. Specimen fixed at end of rotating arm which is angularly accelerated until centripetal acceleration equals desired test accelerations.	Simplicity. Moderate power requirement unless rapid acceleration is attempted. Balanced machine. All motions constrained. Precise acceleration control if motor driven.	Dynamic effects are not simulated, since rate of application of acceleration must be low if excessive accelerations transverse to test axis are to be avoided. Slip rings required for instrumentation.
"Snap-out" Centrifuge. Centrifuge arm is brought to speed with test specimen at center of rotation. Test initiated by moving (or allowing centrifugal force to move) specimen rapidly to test radius.	Machine may be relatively simple if centrifugal force is used to move specimen.	High transverse (Coriolis) accelerations limit rate of build-up if specimen is moved rapidly to test position. Normally an unbalanced machine. Slip rings required for instrumentation.
"Varigee" Centrifuge. Arm is quickly brought to speed by programmed clutching to a preaccelerated motor-driven flywheel. Specimen mounted pendulously on turntable so that axis roughly follows axis of resultant of tangential and radial accelerations during angular acceleration.	Balanced machine. Modest power requirements. More rapid build-up possible than in conventional centrifuge. Rate of termination of acceleration also controllable.	Rate of acceleration build-up limited by torque capacity of clutch and by lag in motion of pendulous turntable in lining up with resultant of accelerations.
"Rotary Accelerator." Arm is accelerated to full angular velocity in 60° of motion by means of single-shot pneumatic arrangement. Cam-programmed turntable keeps specimen axis aligned with resultant.	Rate of acceleration build-up limited only by rigidity of parts. Minimized transverse-acceleration components. All motions constrained and reproducible. Instrumentation may be wired directly to specimen.	Unbalanced machine. Acceleration gradually drops off during test.

THE ROTARY ACCELERATOR PRINCIPLE

The local instantaneous directions and magnitudes of resultant acceleration vectors at various points on the test-specimen turntable during the angular-acceleration phase of motion in the Rotary Accelerator are shown in Figure 1. Magnitudes are shown with respect to the desired test value; directions are with respect to the axis of the test specimen. Initially, the test-specimen axis is perpendicular to the arm, since the acceleration is purely tangential as the arm starts to move. The rate at which the maximum

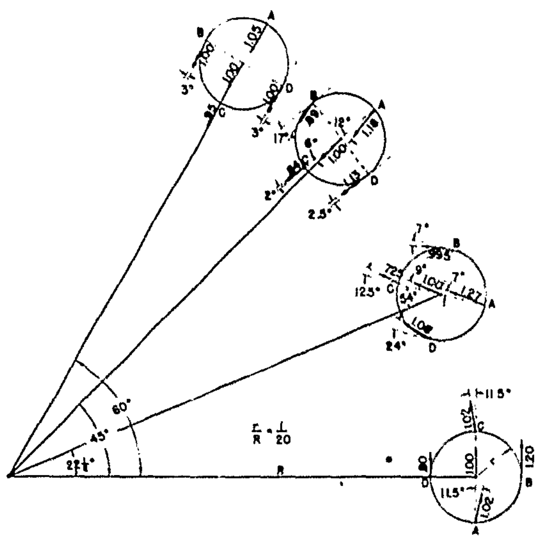


Figure 1 - Instantaneous directions and magnitudes of resultant acceleration vectors during the angular arm-acceleration phase of rotary accelerator motion. Magnitudes are with respect to the desired test value; directions are with respect to the axis of the test specimen.

acceleration is reached is dependent solely on how rapidly mechanical restraint on the arm is removed, and the force of an air cylinder pushing on the arm becomes available to produce angular acceleration of the arm. As the angular velocity of the arm increases, the centripetal component of acceleration increases continuously until at 60° of motion full angular velocity is reached, the arm free-wheels, and the motion is thenceforth that of a conventional centrifuge. Meanwhile, the tangential component has declined to zero because of air expansion in the accelerating cylinder, and a properly designed cam has kept the test-specimen axis aligned as closely as possible with the instantaneous resultant of the two components.

The necessary angular accelerations of the arm and of the turntable with respect to the arm are determined by the solution of certain differential equations of motion given in the previous article,[2] and a modification of the theoretical programming is introduced to bring the turntable smoothly to rest (with respect to the arm) at the end of the angular-acceleration phase. This modification, plus the acceleration gradient common to all centrifuges, results in certain deviations from the ideal of a constant, purely uniaxial acceleration of the test specimen, as shown in Figure 2. These deviations also appear in the acceleration ratios and resultant angles with the specimen axis given in Figure 1. However, a good over-all approximation of the desired pattern results.

ROTARY ACCELERATOR DESIGN

The physical embodiment of the Rotary Accelerator principle shown in Figure 3 is that contemplated in 1949 and described in the previous article. During design of an actual machine in 1950-51, major changes and simplifications resulted in the arrangement shown in Figure 4; but no changes in the fundamental motions or theory of operation were made. The following paragraphs describe the reasons behind the design changes and, together with the previous article, constitute a current description of operation. Dimensions and operating characteristics of the machine in its two alternate assemblies are given in Table 2.

In the original design, a dual balanced system was contemplated; but as design progressed, evidence developed which showed the essentialness of mechanical construction, strong enough to avoid disaster in case of failure of a large test specimen or in the event one of the two air cylinders providing the power should get out of step or fail to fire. It was therefore a short step to

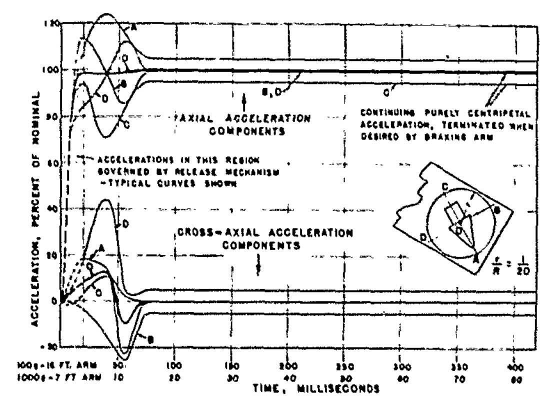


Figure 2 - Theoretical acceleration-time curves in the direction of the test axis (axial) and transverse to it (cross-axial) for the modified programming used in the rotary accelerator. Locations are equivalent to those on a 5.4-in. diameter circle for the small arm and a 12-in. circle for the large arm of the actual machine.

[2]See footnote 1.

TABLE 2
Rotary Accelerator Dimensions and Characteristics

Design Considerations	Small Arm	Large Arm
Diameter (to turntable center)	9 ft	20 ft
Approximate maximum acceleration	550 g's	75 g's
Approximate maximum peripheral velocity	282 fps	94 fps
Approximate maximum angular velocity	600 rpm	149 rpm
Test specimen weight at rated acceleration	8 lb	100 lb
Test specimen space: Height Diameter	5 in. 12 in.	29 in. 29 in.
Acceleration build-up time	5-35 ms	10-50 ms
Cylinder bore and stroke	6 x 27 in.	6 x 27 in.
Cylinder pressure	3000 psi	3000 psi
Piston thrust	75,000 lb	75,000 lb
Accelerating torque	168,750 lb-ft	168,750 lb-ft

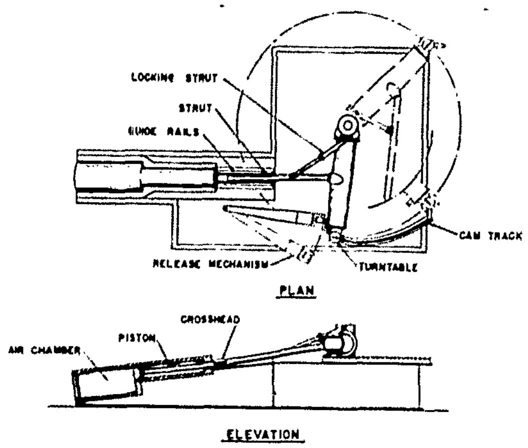


Figure 4 - Plan and elevation of Rotary Accelerator as constructed, showing small (54-in. radius) arm. Arrows on turntable indicate axis of test specimen.

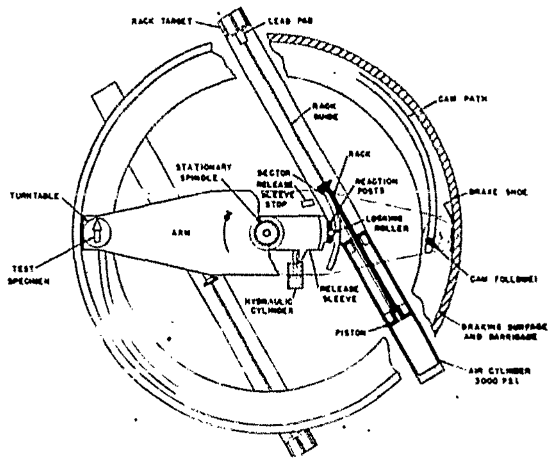


Figure 3 - Schematic view of original plan of Rotary Accelerator

provide a foundation and arm bearing arrangement of sufficient strength to resist the entire centrifugal load and thus eliminate the second turntable, arm, cylinder, etc. In the new design, large roller bearings at the arm hub carry the unbalance and the machine foundation bolts to a welded grid of 12-in. I-beams set in a 24-in. concrete slab cast integral with the basement floor of the Ordnance Environmental Laboratory at NOL.

A rack and sector arrangement, for translating the rectilinear piston movement to rotation of the arm, was abandoned in favor of a strut or connecting rod. At the end of the power stroke the piston and piston rod are stopped by compression of air in the forward end of the air cylinder, while the strut, which has been resting in a socket at the end of the crosshead, remains attached to the arm

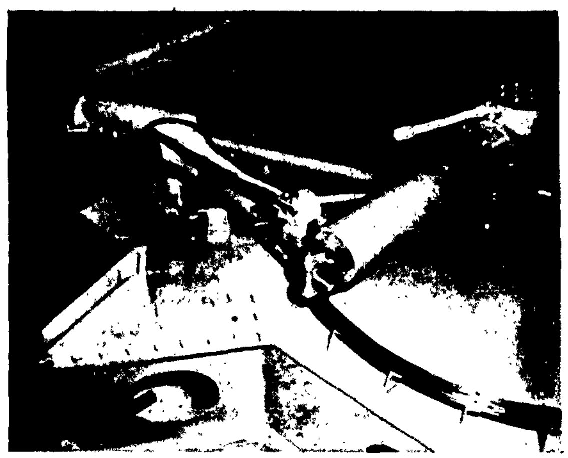


Figure 5 - View of Rotary Accelerator prior to firing. Test specimen (fuze) on turntable aligned with initial acceleration perpendicular to arm. Air cylinder in left background and hub (right background) remain in place in alternate assembly with 10-ft radius arm. Release mechanism (center left) is moved out to match end of longer arm, and cam track (right foreground) is superseded by track at greater radius. Release mechanism bumper (left foreground) is removed.

and is held in by the locking strut. This arrangement prevents centrifugal force from flinging the strut outward to flail the stationary parts of the machine.

Consideration of the effects of changes in peak acceleration attainable with various arm radii for a fixed test-specimen weight and air pressure showed that little reduction in acceleration occurred up to 4-1/2-ft radius for an 8-lb test load. And since the larger the radius in comparison to test-specimen size the smaller the acceleration errors at the specimen extremes, the arm radii were increased to 4-1/2 and 10 ft.

It was considered advisable to restrain the arm prior to release in such a manner that the strain distribution before and after release would be generally similar, thus minimizing the tendency for release to set the arm into vibration. Accordingly, the release mechanism is mounted on a pivot just outside the path of the arm and restrains the arm through a link attached to the arm just inside the turntable (Figure 5). Thus, the bending load on the arm before release (due to the cylinder force balanced by the restraining force on the link) is almost identical to that immediately following release (when the cylinder force is balanced by inertia force, whose centroid is near the link location). A cam arrangement, readily visible in Figure 6, drives the release mechanism safely out of the way of the arm before the completion of the first revolution.

A similar problem was to provide clearance for the cylinder on subsequent revolutions and still have the driving force act through the torsional axis of the arm to avoid the need for excessively heavy arm construction. This problem was solved by inclining the cylinder axis as shown in the elevation view of Figure 4. Such design also minimizes torsional vibration excitation of the arm at the time of release.

By selecting the proper air chamber volume and ratio between initial air pressures ahead of and behind the piston, it was possible to achieve the necessary piston travel-air pressure relationship without venting any air. With 3000 psi behind the piston, approximately 600-psi "muzzle" pressure is used to stop the piston safely and to provide the correct tangential acceleration at each point of piston travel as the arm is brought up to speed.

The release mechanism provides control over the rate of application of the test acceleration by governing the rate at which the arm is released

Figure 6 - Rotary Accelerator "in flight" at about 155° of rotation from start. Strut has separated from piston rod and is retained by locking strut attaching it to hub. Turntable has rotated 90° from its initial position with respect to arm and is latched in place. Cam on hub beneath arm is pushing release mechanism out of path of arm. Maximum of about 8000 hp is released during brief period of arm acceleration.

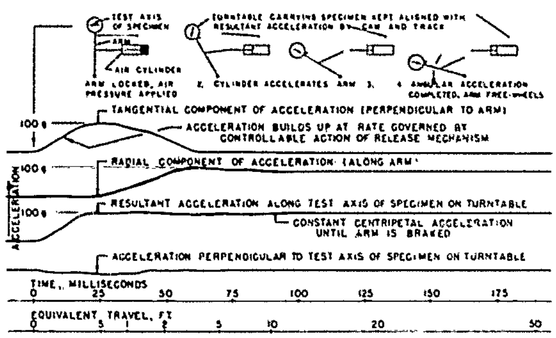


Figure 7 - Rotary Accelerator acceleration-time oscillograms at 100 g's with high retarder pressure (slow acceleration build-up). A similar shot with zero retarder pressure would reach peak in about 6 ms.

and the rate at which the cylinder force becomes effective in accelerating the arm. When a fork restraining a pair of rollers is withdrawn by means of a hydraulic cylinder, the rollers release the link attached to the arm. Two "retarder" cams also must be forced out of the way by the link as it starts to move. These cams are loaded by small air cylinders whose pressure is controlled independently of the piston pressure which

accelerates the arm. The higher the retarder pressure, the slower the acceleration build-up for a given test acceleration.

Acceleration is terminated by an air-actuated band brake on the main shaft. The arm is brought to rest in about one revolution, resulting in a transverse acceleration component, during braking, of approximately 10 percent of the test acceleration. No automatic control of braking has been fitted since, to date, manual control has proved satisfactory for test needs. Should short-duration or programmed-acceleration-decrease tests be needed, an automatic control may be necessary.

ROTARY ACCELERATOR PERFORMANCE

Acceleration-time curves measured on the machine are shown in Figure 7. The top two traces, taken with the turntable disconnected from the cam track, show the tangential and radial components which the turntable motion combines in a regular test to produce the desired test-axis acceleration (third trace). The undesired transverse acceleration is shown in the fourth trace. As predicted by theory, this acceleration is sufficiently small to satisfy conditions for most test problems.

A lower-speed record of a complete acceleration test appears in Figure 8. Scales converting the time scale into equivalent distance and velocity values (assuming accelerated motion from rest) are given. This test, an extremely long one for practical purposes, was conducted to illustrate the drop-off in acceleration of about 6 percent per second which occurs during free wheeling. This drop-off is primarily due to friction rather than windage, and has shown some tendency to decrease in subsequent tests. It may possibly be a limitation in certain tests.

Over 400 test firings have been made to date. As the records show, the Rotary Accelerator produces the intended smooth acceleration pulses free from extraneous transverse accelerations not theoretically present, and free from evidence of arm-frequency excitations. Operation of the machine is rapid (less than 5 minutes from one shot to the succeeding one if the test specimen does not have to be reoriented or an instrumentation set-up revised) and involves no expendable parts. To date, calibration has been carried up to only 150 g's, because test problems have not required higher values. The alternative 10-ft radius arm assembly for handling large items at lower accelerations has been completed but has not been assembled.

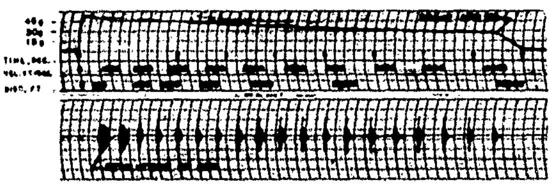


Figure 8 - Acceleration-time record of complete test. Acceleration drop-off in 8 sec is approximately 48 percent of initial value. Equivalent travel from rest during acceleration equals 53,000 ft. Data were recorded with Statham pick-up and Brush recorder.

Problems on which the Rotary Accelerator has been used are listed below. In cases where laboratory results have been checked by subsequent field firings, direct correlation has been found.

1. Determination of minimum arming accelerations and distances for inertia-armed rocket fuzes.
2. Investigations of "hang-ups" in missile-timing mechanisms during boost.
3. Investigation of performance of set-back unlocking arrangements for rocket fuzes.
4. Determination of calibration factors for integrating accelerometers.

From a practical standpoint, a cardinal virtue of the machine is that a complete test normally involves only from three to ten revolutions. Semipermanent twist-up cables can thus be used for instrumentation, allowing complete freedom in choice of end instruments and evasion of the slip-ring problem.

DISCUSSION

B. K. Wernicke, WADC: What is the maximum size and weight of the test specimen which can be checked with this equipment?

Armstrong: With the short arm, we can accommodate specimens up to about 12 in. in diameter and 5 in. high. The machine is designed to get full acceleration of 500-600 g's with an 8-lb load. The large arm of 10-ft radius has room for equipments up to 29 in. in diameter and 29 in. high. The maximum acceleration is about 75 g's with a 100-lb load. We can get correspondingly higher acceleration if the load is smaller.

Wernicke: Do you know of any testing machine that takes 100- to 200-lb equipment for testing the relative loading up to 10 g's extended acceleration?

Armstrong: This machine can handle that. You could put several hundred pounds on it - as many as there are room for, provided you want a lower g.

Wernicke: Is the size limitation of the equipment about 29 in.?

Armstrong: Twenty-nine inches in diameter and 29 in. long. In the first test we ran, however, we had a specimen outside the limit, and it was necessary to saw off a few inches.

Wernicke: We had a discussion about this question of size and weight. We put out a specification for test equipment weighing about 250 lb for prolonged acceleration at a lower g value than the maximum value given for the Rotary Accelerator. We were informed that there was no testing machine available which could take this size of specimen.

Armstrong: This machine, of course, is not required if you have no rapid build-up time requirement on the acceleration. Conventional centrifuges would perform the same function, although centrifuges of the size required are not common.

M. F. Busch, Naval Gun Factory: You mentioned the fact that you put the gun on an angle horizontal to the table and, also, the fact that it had some recoil. Will you please explain why this method of mounting the gun is desirable?

Armstrong: The question may be stated as follows: What does the inclination of the axis of the cylinder have to do with torsion in the arm?

We want to have as many of the forces as possible act through the torsional axis of the arm, because we have a system accelerating at a very high rate. We can release the arm in as little as five ms, and any unbalanced forces there would tend to set the arm into torsional vibration or bending vibration.

We have calibrated the accelerator only up to a little over 100 g's, because our tests have been in that range. So far, there have been no indications of any excitation. We would like to think that this is because of our forethought in design. Maybe excitation would be present if we pushed off to one side. We are trying to reduce what you might call the coefficient of b-o-i-n-g that you get if it pushed too far off center.

* * *

THEORY AND DESIGN OF AERIAL CAMERA VIBRATION ISOLATION AND STABILIZATION

B. K. Wernicke, WADC

The design of a steady platform eliminating loss of resolution of the image caused by airplane movement and vibration is discussed. The essential dynamic requirements of a stabilized system are analyzed.

INTRODUCTION

The problem of steadying an aerial camera in the aircraft is an important task, even though it is a small section of the art of aerial photography and aerial reconnaissance. Some of the requirements discussed here have been considered before, but a complete system adequate for military application has never been engineered. The engineering of a center of gravity camera mount system has been carried on in the last few years to provide the means for a guaranteed steadiness under all practical airplane conditions (Reference 1).

The discussed aerial camera mounting system will eliminate to the maximum extent the detrimental effect of aircraft acceleration. The essence of the system is to steady a camera in an airplane within the limits which are required to exploit fully the optical qualities of the camera and of the sensitized film material, for any focal length chosen according to tactical viewpoints, and for extremely long exposure times up to 1/4 sec.

The system provides the basis for proper compensation of the image motion due to the forward motion of the airplane, thus maintaining the full steadiness of the image in reference to the film. This quality of steadiness also is guaranteed under military conditions.

By means of the technique of servo control, the system provides the energy needed to withstand the disturbing forces encountered in use.

GENERAL PERFORMANCE REQUIREMENTS

The general performance requirements are determined by the aircraft conditions and the photographic requirements.

We may distinguish between three different modes of aircraft acceleration:

1. Low frequency oscillations of the aircraft determined by the flight dynamics of the airplane;

2. High frequency oscillations of the airplane structure, called vibrations;

3. Shock, sudden accelerations of high magnitude and of short duration in one direction.

Stabilized mounts used during the past years eliminate mainly the low frequency oscillations of the airplane around the roll and pitch axes. The camera is kept in a defined direction with an accuracy which is of the order of minutes of arc. However, these mounts do not eliminate the

deteriorating effects of aircraft vibrations on the photographic quality if long exposure times are needed. Long exposure times and the application of long focal length cameras require a higher quality of steadiness in order to exploit the possibilities of obtaining better information by means of longer focal length. These requirements introduced the problem of the steadying of cameras within a few seconds of arc during exposure times up to 1/4 sec.

The required degree of steadiness is related directly to the camera focal length and exposure time. The longer the focal length and exposure time, the higher the required degree of steadying. The following table shows the steadying errors that may be tolerated.

Focal Length (in.)	Steadying Error (Movements During Exposure) in sec of arc	Lines per mm for Which the Steadying Error is Equal to the Distance Between Lines
6	24	28
12	12	28
24	6	28
48	3	28

Fortunately the camera is extremely sensitive to rotational movements only. Pure translatory movements, of magnitudes occuring during flight, do not influence the image quality to a measurable degree. However, in order that the camera be steadied to the extent that the quality of the optical system be fully utilized, a special arrangement has to be provided which filters the influence of the aircraft vibrations to an extremely high degree.

There have been numerous attempts to isolate the camera from airplane vibrations and shock by means of standard vibration isolators, but without consistent success because of the high degree of steadiness required compared with the magnitude of the disturbing forces. Without very sensitive control means, the accuracy of the CG position within the center of the isolator spring system is insufficient, and the inertia of the camera system is too low compared to the excitation forces of the aircraft structure. Since no steady reference is provided, the inertia of the camera would have to be increased until sufficient steadiness is accomplished. This means that, in most of the conditions experienced, an enormous increase in weight and volume is not desirable for this type of equipment.

DESIGN REQUIREMENTS

A thorough study of those factors which have to be controlled leads to the so called CG mounting system with specific design and control features.

Figure 1 shows one of the first experimental items which actually fulfills the requirements.

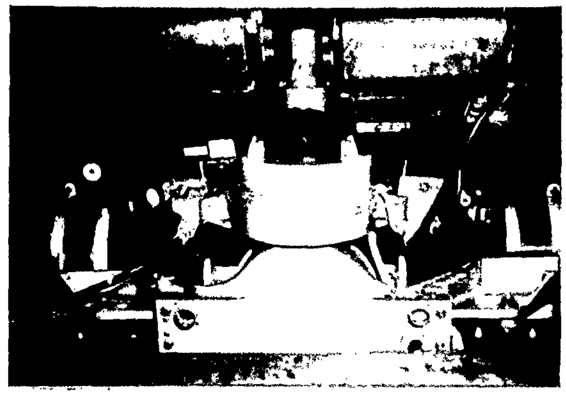


Figure 1 - CG torquer mount

This system is distinguished by the following design features:

1. The steadied member consisting of camera, gimbal, and steady reference unit is supported by two gimbal axes having a bearing friction as low as possible. (The two axes are transverse to each other.)

2. A steady reference unit consisting of a high-precision gyro with a proper electrical signal output for servo control is provided on the steadied member.

3. Torque motors are provided at each gimbal axis. The torque supplied by these motors is controlled by means of an amplified transmitter signal of the steady reference mounted on the steadied member.

A number of specific design requirements must be fulfilled to the highest possible degree in order that a maximum steadiness be obtained. These requirements may be distinguished by two groups:

1. Requirements or measures necessary to eliminate the factors disturbing the steadiness.
2. The means necessary to provide a steadying torque enforcing the desired steadiness of the system.

It is a necessity to reduce the disturbing forces in order to lower the power consumption as much as possible.

Provisions to eliminate those factors disturbing the steadiness are:

1. CG mounting

   The camera must be supported in the center of gravity by the axes of suspension which must provide the full rotational freedom in all those coordinates in which the camera is to be steadied.

2. Elimination of a gear train

   No gear train transmission can be introduced between the steadied member and the torque-supplying servo components connected to the oscillating frame of the airplane.

3. Reduction of bearing friction

   The friction in the bearings of the supporting axes must be a minimum as no friction torque is desirable.

4. Maximum flexibility of electrical cables

   Any cable connections between the steadied camera and the airplane must be especially designed to introduce a minimum of disturbing torque to the steadied camera.

5. Any device for measuring the angular movements between the steadied member and the airplane must have a minimum of inertia and friction.

6. The weight shift by film transport must be corrected by an automatic weight control, which shifts a movable weight in accordance with the film shift to balance the system. (Residual unbalance increases the necessary gain of the servo loop and the absolute amount of servo torque required to steady the camera and to compensate for the disturbing torques.)

7. Any other disturbing torque between the steadied camera and the airplane frame is to be avoided or must be insignificant compared to the steadying servo torque.

Means are required to provide a steadying torque, enforcing the inherent steadiness. A steady reference unit attached to the camera delivers a transmitter signal as a function of the camera deflection from the steady reference line. Normally this reference will be provided by means of a horizon, rate, or rate-integral gyro of proper accuracy.

A torque, derived from the transmitter signal, is fed into a proper torque device that is active between the steadied member and structure. This device may be of electrical, pneumatic, hydraulic, or miscellaneous nature. The experimental items are electrically operated.

The torque device must fulfill the following characteristics:

1. A threshold sensitivity equivalent to 1 to 3 sec of arc deflection of the camera from steady reference and extremely smooth operation must be accomplished.

2. The servo torque must be a function only of the one reference signal and must not be distorted by position, angular velocity, or angular acceleration of the aircraft.

3. The gain of the servo torque, that is the torque produced per unit deflection of the camera from the reference line, must be sufficiently high in order to suppress all factors disturbing the steadiness. (See dynamic requirements.)

4. The maximum control torque at saturation must be higher than the sum of all possible disturbing factors.

Every practical mount design will have certain limitations in the reduction of forces disturbing the steadiness. The following are estimated deviations of the design from perfect conditions:

1. Depending on design and protection against environmental conditions, the bearing friction of the gimbal axes may vary from 0.5 to 2 lb-in. for a 100-lb camera and 2 to 5 lb-in. for a 300-lb camera.

2. The remaining unbalance to be encountered is estimated as 2 to 10 lb-in. for cameras of 50 to 300 lb. A certain amount of unbalance will be unavoidable if the whole system cannot be sealed completely against balance-disturbing influences. Since the camera magazine has to be changed, the possibilities of sealing the unit are very limited. Even if the weight shift of the film is compensated, it will be safer in service to provide the servo control for sufficient unbalance to cover the remaining unbalance

which may be encountered under rough military conditions. This is estimated to be from 5 to 10 lb-in. for cameras of 50 to 200 lb.

3. The weight shift caused by the film is of the order of 30 to 80 lb-in.

4. Cable elasticity can be 10 lb-in. and more. It is advisable to connect the airplane cable to the outer frame of the mounts and to provide a special cable inside the mount gimbal in such a manner that the torque applied does not vary much.

DYNAMIC REQUIREMENTS

A scheme of the dynamic conditions of "torquer mount systems" is shown in Figure 2 representing only one axis. The center point Z is the axis of suspension. CG is the center of gravity of the camera and gimbal creating an unbalance

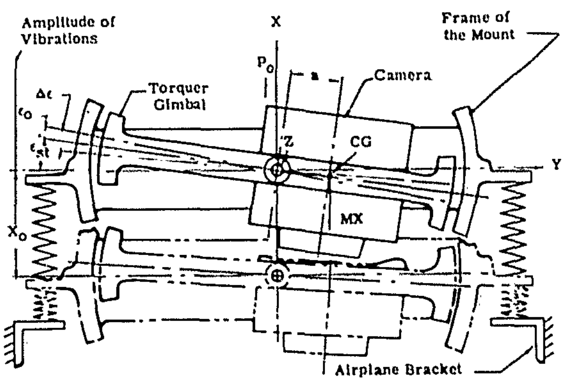


Figure 2 - Schematic of a torquer system camera, CG-mounted

Z = gimbal axis
CG = center of gravity of camera and gimbal
G = weight of camera and gimbal
T_u = aG = unbalance around Z-axis
K = steepness factor of servo-controlled torque
ϵ_{st} = servo angle = T_u/K = static deflection caused by the unbalance (T_u)
ϵ_o = amplitude of rotational vibrations
$\Delta\epsilon = 2\epsilon_o$ = steadying error caused by forced vibrations on step input accelerations
$T_s = K(\epsilon_{st} + \dot{\epsilon})$ = servo controlled torque
X_o = amplitude of translatory vibrations of the gimbal axis (Z) in the X-direction

(T_u = aC) around the Z axis. The torquer gimbal contains electromotor coils producing a torque controlled by a steady reference unit attached to the camera.

A quantitative dynamic analysis has been made of the rotational vibrations of a solid camera body caused by translatory vibrational forces. Figure 3 shows a result of the analysis, a diagram of the required gain of the torquer system in lb-in. per millirad as a function of the remaining unbalance. The parameters are the following:

1. $\Delta\epsilon$ = maximum steadying error = 3 sec of arc = 0.0145 millirad
2. F_{tr} = translatory forced vibration frequency of the pivot bearing = 7 cps
   W = $2\pi F_{tr}$
3. X_o = forced vibration amplitude of the pivot bearing = 0.004 in.
4. Tolerated angular vibration amplitude of the steady member = 3 sec of arc
5. I = inertia of camera in suspension point
   = 0.5 slug-ft^2 in Curves a_1, a_1', b, and c
   = 1.0 slug-ft^2 in Curves a_2 and a_2'
   = 2.5 slug-ft^2 in Curve a_3
6. δ = damping

Formula for Curves a_1, a_2, a_3, a_1', and a_2':

$\delta = 0$

$$K = \frac{12IW^2}{1000} \pm \frac{2X_oW^2T_u}{g\Delta\epsilon}$$

Where K = gain of servo torque
g = 386 in./sec^2

Formula for Curve b:

$\delta = 0.5$ critical

$$K = \frac{12IW^2}{2000} \pm \sqrt{\left(\frac{2X_oW^2T_u}{g\Delta\epsilon}\right)^2 - 3\,\frac{12IW}{2000}}$$

Formula for Curve c:

$\delta = 1.0$ critical

$$K = -\frac{12IW^2}{1000} + \frac{2X_oW^2T_u}{g\Delta\epsilon}$$

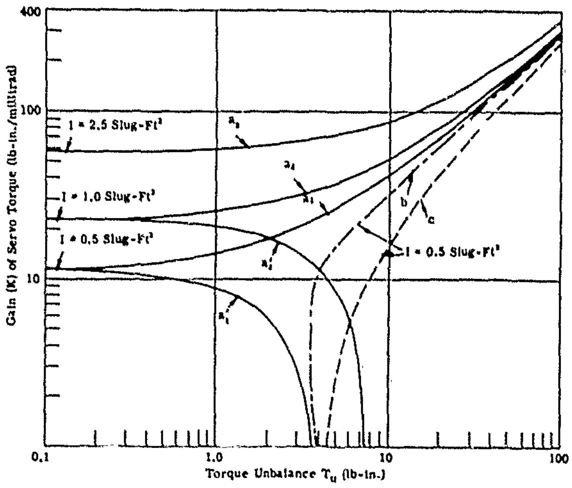


Figure 3 - Graph of the required gain of the servo torque as a function of the unbalance (T_u)

It is easy to see from Figure 3 that an incorrectly designed steadying torquer mount system could result in an amplification of the vibrational amplitude if the gain provided in the servo system were not adapted properly to the conditions encountered in the aircraft. For example, Curves a_1' and a_2' cross the abscissa at points 3.6 and 7.2 lb-in. tolerable unbalance under the condition that the steadying error does not exceed 3 sec of arc. This means furthermore that the inertia of the system is sufficient to take care of this amount of unbalance. Following Curves a_1 or a_2, the tolerable unbalance (T_U) decreases and becomes zero at gains of about 12 and 22 lb-in. per millirad respectively. This point of the ordinate represents the resonance conditions between the frequency of the forced vibrations of the support and the natural frequency of the mount servo system including the inertia of the camera-mount combination. If the natural frequency of the mount servo system is close to the natural frequency of the vibration introduced through the support, the camera may oscillate with extremely high amplitudes.

In order that the torquer system provide an essential steadying effect against unbalance, the gain must be high enough to provide the mount servo with a natural frequency which is higher than the frequency of the airplane vibrations of the pivot introduced through the isolated support. For this reason, extremely low vibration isolators are used to eliminate all high frequencies of the airplane frequency spectrum.

DAMPING

The advantage of high damping is shown by Curve c, Figure 3. A high amount of damping eliminates completely the possibility of critical conditions and amplitude amplification and results in a vibration reduction for any gain condition. The damping can never be too high. If it consists strictly of error-rate damping, no angle of lag can appear. In order that the design shall rely on a high damping factor, an extremely clean damping circuit must be provided.

Figure 4 shows in the most common form, the reduction of the vibration amplitudes as function of the ratio of excitation frequency to servo frequency, and the effect of the damping in the resonance area. The experimental aerial camera mount (Figure 1) fulfills most of the theoretical requirements considered essential for steadying.

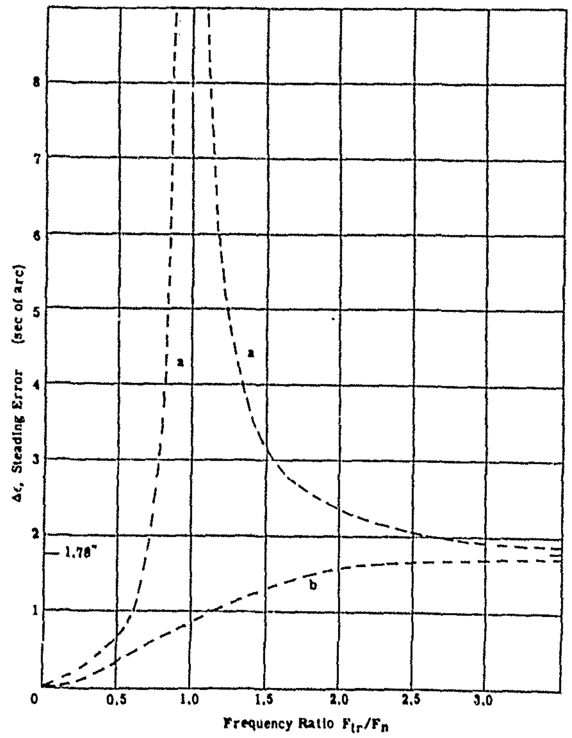


Figure 4 - Steadying error (Δε) as a function of the ratio of the forced frequency of the vibrating support to the natural frequency of the servo system of the torquer mount

The parameters are the following:

1. F_{tr} = frequency of the support
   $W = 2\pi F_{tr}$

2. T_u = unbalance
   = 1 lb-in.

3. X_0 = vibration amplitude of the support
   = 0.01 in.

4. F_n = natural frequency of the mount system
   $W_n = 2\pi F_n$
   $\frac{F_{tr}}{F_n} = \frac{W}{W_n}$

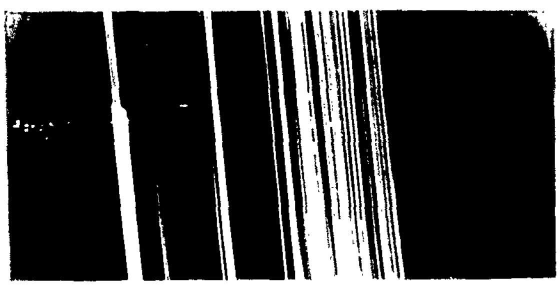
5. I_z = inertia
   = 0.5 slug-ft^2

6. δ = damping
   = 0 for Curve a
   = 1.0 critical for Curve b

Formula for Figure 4:

$$\Delta\epsilon = \frac{2X_0W^2T_u}{gI_z \; (W_n^2 - W^2)^2 + 4\delta^2W^2}$$

An example of typical flight results is illustrated in Figure 5. The lines are traces of stationary ground lights, recorded by an exposure of 9 sec in a K-36 camera of 24-in. focal length. The straightness over a period of 9 sec (Figure 5a) indicates that an extremely high steadiness under the influence of the airplane motions is accomplished. Compare Figure 5a with Figure 5b - the nonsteadied image which was taken a minute later with the mount gimbals caged to the aircraft. Deflections from a straight line, during an exposure time of 0.1 second, do not exceed 3 seconds of arc at any point of the recording (Figure 5a). In this particular case, the high-frequency vibration amplitudes are almost zero by the effect of the high inertia, since the system is in a well-balanced condition.



a. Mount on

b., Mount off

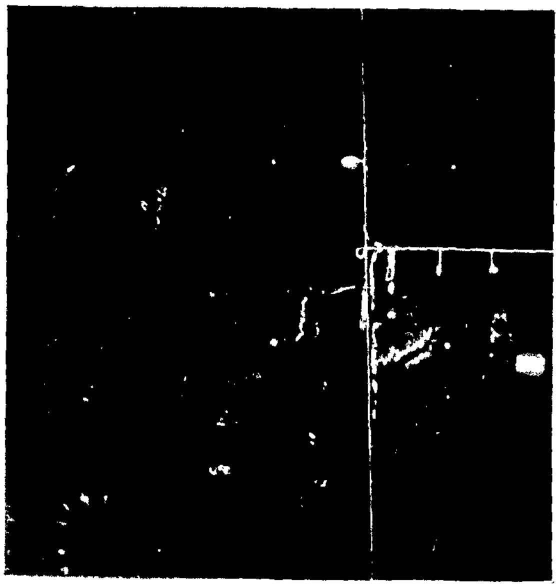
Figure 5 - Recordings of camera motion in flight

Figure 6 - Night photo--rigid mounting (standard isolator)

Figures 6 and 7 illustrate the difference in information between a nonsteadied and a steadied picture, from night photos taken at 10,000 ft in August 1952. An experimental steadied torquer mount was used.

CONCLUSION

A great variety of aerial cameras are in use in the Air Force. The accuracy requirements vary with the type of cameras and their applications. Dynamic considerations and experimental results show that it is possible to obtain the required steadiness with a reasonable amount of effort. The requirements of CG mounting necessitate a specific design for each category of



cameras and camera combinations. The technical means applied to the task are of considerable influence on the simplicity, the reliability, the unique character of the design, and the cost of equipment. A clean signal and/or proper filter networks for the servo loop are of major importance for the accomplishment of the required sensitivity.

Figure 7 - Night photo--torquer mount

ACKNOWLEDGMENTS

Acknowledgment should be given to the personnel of the Wright Field Photo Reconnaissance Laboratory, and in particular to Mr. Mungall, Chief of Aerial Equipment Branch, and Dr. Heidelauf and Dr. Mestwerdt, consultants of the Photo Reconnaissance Laboratory.

REFERENCE

1. Wernicke, B. K., Stabilization of cameras to eliminate loss of resolution by airplane movement and vibration - summary and theory of essential requirements, AF Technical Report 6316, April 1951

DISCUSSION

Leon Wallerstein, Jr., Lord Mfg. Co.: I understand that there is considerable emphasis placed on getting washers that are relatively friction-free, for gimbal supports.

Wernicke: Yes.

Wallerstein: Do you have a servo torque drive to steady the camera and compensate for the disturbing torques?

Wernicke: Yes.

Wallerstein: Is the purpose of the low frequency simply to reduce the torque necessary to enforce steadiness, or does it have some other significance in the suspension?

Wernicke: Reduction of the frequency is necessary only in order to reduce the power and the torque required to enforce the steadiness. If you have no frequency effect and no unbalance, you could make an inertia steady system that would remain in perfect balance for the time of exposure.

* * *

A PORTABLE 100-FOOT DROP TESTER*

J. C. New, NOL

A portable low-cost shock-cord-actuated drop tester with striking velocities equivalent to 100 ft free-fall is described. This 2500-lb machine occupies only 15 sq ft of floor space, is 11 ft tall, and has a total floor loading of 200 psf. A unique stopping device varies the acceleration pulse up to 500 g's with time duration from 10 to 25 ms.

INTRODUCTION

The machine to be described has been popularly called "A Portable 100-ft Drop Tester." The 100-ft designation comes from the fact that the maximum velocity change experienced by the carriage of this machine is equivalent to a 100-ft free-fall drop under gravity acceleration only. The Navy has officially designated this machine as the "Drop Shock Tester Mk 7 Mod 0."

This shock testing device was designed by staff members of the Naval Ordnance Laboratory to meet the needs of ordnance contractors. It is intended for use in industrial plants where elaborate shock testing facilities are, in general, not available. This machine is intended to provide a simple and direct method of screening complex components for shock resistance and to detect manufacturing defects. Although the Mk 7 Mod 0 Shock Tester was designed to simulate a service-type shock for certain types of naval ordnance, it is believed that simple modifications of the machine can be made to extend its usefulness to other shock problems. In performing a complex-item type of inspection on ordnance for manufacturing defects, it is necessary that shock tests be judiciously applied through practical testing devices at minimum cost. This is the goal of the subject drop tester.

DESIGN SPECIFICATIONS

The following design specifications are established for this drop tester.

1. The machine must be low-cost, portable, easy to install in an industrial plant, and have a floor loading in the range of 200 to 250 psf, yet occupy a minimum of floor space. It must use standard, commercially available parts wherever possible and be simple and easy to operate.

2. A test load of 10 lb must experience a deceleration for not less than 10 ms with a peak of 500 g's. A maximum test load of 25 lb must experience a deceleration for not less than 10 ms with a peak acceleration of 125 g's.

3. The space available for mounting the test specimen must be not less than 8 in. diameter by 8 in. high.

MECHANICAL DETAILS

How well these design goals have been met can be appraised from Table 1.

TABLE 1

Floor space	$2\frac{1}{2}$ ft x 6 ft
Height	11 ft
Weight	2500 lb
Floor loading (including dynamic)	200 psf
Test specimen space	8 in. diam. x 8 in. high
Test load at maximum shock parameters	10 lb
Maximum test load	25 lb
Propulsion system	5/8-in. Elastic shock cord 3/16-in. Wire rope tow cables
Maximum available energy	3300 ft-lb
Gear reductor hoist unit	150:1 Reduction 1 hp, 110/220 volts ac
List of drawings	BuOrd LD 290962

*This machine was exhibited at the 20th Shock and Vibration Symposium

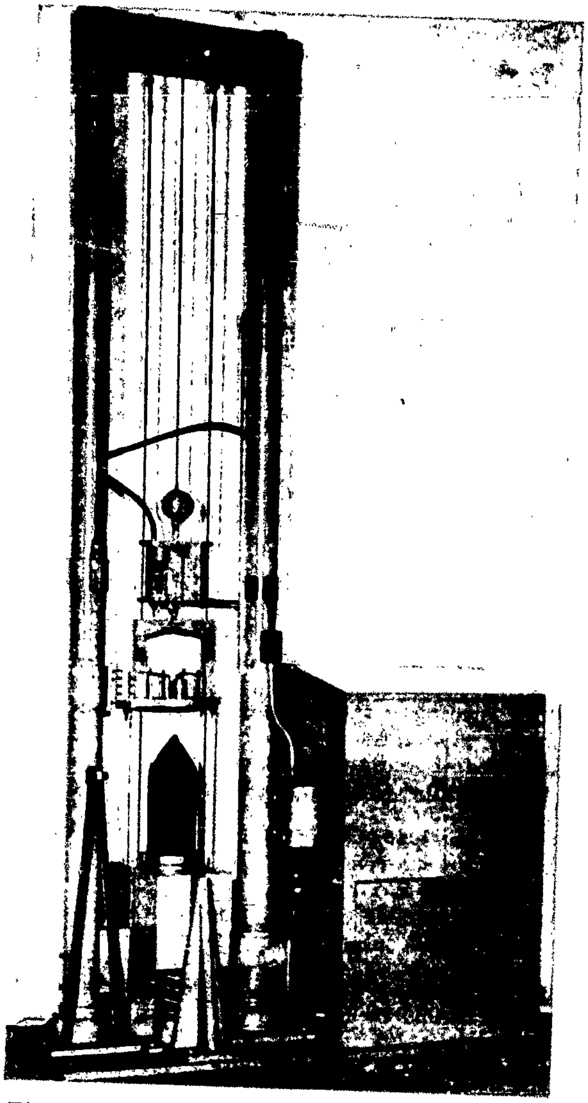


Figure 1 - General view of drop shock tester Mk 7 Mod 0

A general view of the Drop Shock Tester Mk 7 Mod 0 is presented in Figure 1 and the various parts of this machine are identified in the schematic drawing shown in Figure 2.

The essential components of this machine may be identified as follows:

1. Structural framework
2. Test carriage and wire rope guides
3. Hoist and release mechanism
4. Shock-cord propulsion system
5. Stopping device on seismic anvil

In the operation of the machine, the specimen is mounted in the test carriage to which the release mechanism is attached. The carriage is elevated, to the height selected on the drop height scale, by the electrical motor driving the hoisting unit through the gear reductor. As the test carriage is raised, the elastic shock chord located n the left-hand column is stretched by means of the two wire-rope tow cables which are attached, one end to the test carriage and the other end to the shock-cord housing. The test carriage is released by pressing the hydraulic release mechanism actuator. At this instant the carriage, being free in space, is accelerated downward by the force of the stretched elastic shock cord. The carriage is decelerated by means of the stopping device mounted on the seismic anvil, thus producing the desired shock pattern.

The structural framework of this machine consists of commercial 4-in. standard pipe and fittings, and a 2-1/2 ft x 6 ft steel deck plate 1 in. thick which distributes the total dead load of 2500 lb over 15 sq ft.

The test carriage, as shown in more detail in Figure 3, is made of aluminum alloy for light weight. Its rigidity is increased by a steel base plate bolted to the bottom. The carriage has a natural frequency of about 1000 cps, and can accommodate test specimens up to 8 in. diameter x 8 in. high.

The mechanical-type release mechanism is shown in the "released" position in Figure 3 and in the "locked" position in Figure 4. This

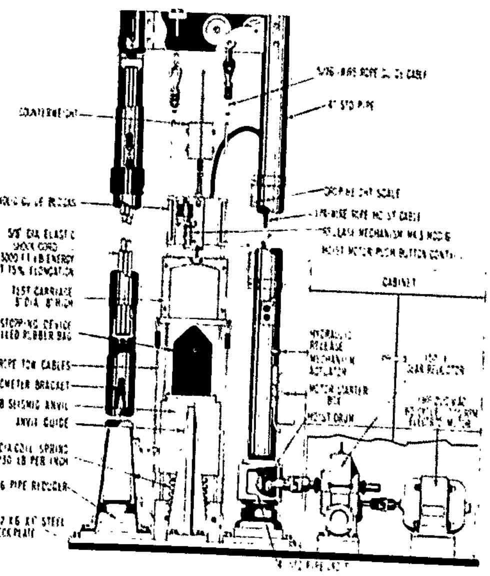


Figure 2 - Schematic of drop shock tester

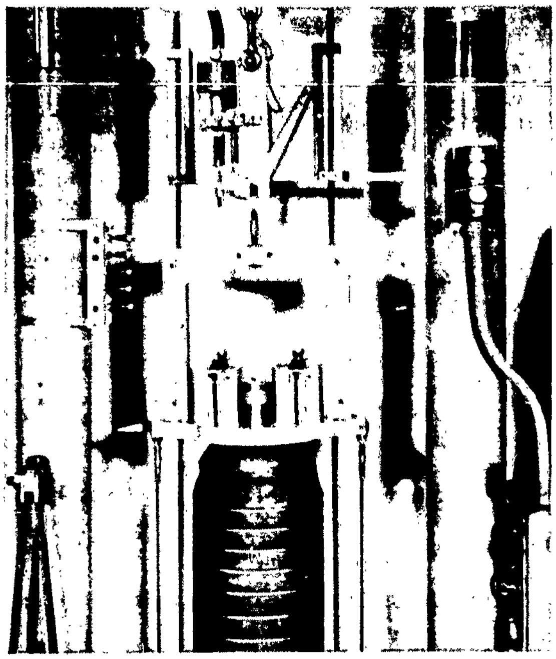


Figure 3 - Release mechanism in "released" position

mechanism is a well-proven standard Navy design which has been modified for hydraulic actuation in this particular application. This modification was made to eliminate any lateral forces during release and to provide more safety and convenience for the operator. Remote operation can be effected when necessary. The build-up time on the release acceleration is on the order of 3 to 5 ms.

The three preceding items presented conventional mechanical engineering problems and demanded very little in the way of specialized experience or research. However, the next two problems—the propulsion system and the stopping device—were more challenging and demanded more experimental research. The design requirements dictated some sort of propulsion system to provide a velocity change of 80 fps. Since the machine was to be kept portable, the acceleration distance had to be kept small and within the floor-to-ceiling distance of the conventional industrial plant, or about 10 to 12 ft. Experience with similar accelerators indicated two sources of propulsion energy—compressed air or elastic shock cord. The former was ruled out because it unnecessarily complicated the entire design.

Elastic shock cord is nothing more than a package of single-strand, rubber bands bundled together with a fabric covering. It has found applications from simple muscular exercisers to landing arrestors of carrier aircraft. Its most appealing attribute in the present application, excepting its simplicity and low cost, was the "shape" of its force-elongation curve as shown in Figure 5. Note, in this figure, that the load has been plotted in percent of the load at 100 percent elongation. There are three parts of this curve that are of interest. First, the load rises very sharply to 30 percent in the first 10 percent of elongation. Second, from this point to 75 percent elongation, the load increases linearly at a much smaller rate. Third, from 75 percent elongation upward, the load increases sharply again and tends to become nonlinear as 100 percent elongation is approached.

The significant point is that the area under this curve is indicative of the propelling energy whereas the load itself controls the initial acceleration of the test carriage. In a design of this type you want the former to be a maximum and the latter to be a minimum. Compromising these

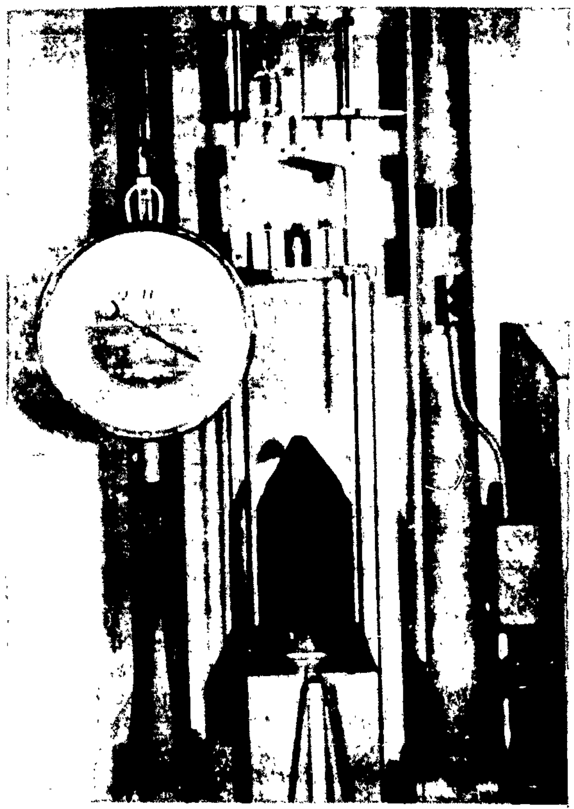


Figure 4 - Release mechanism in "locked" position

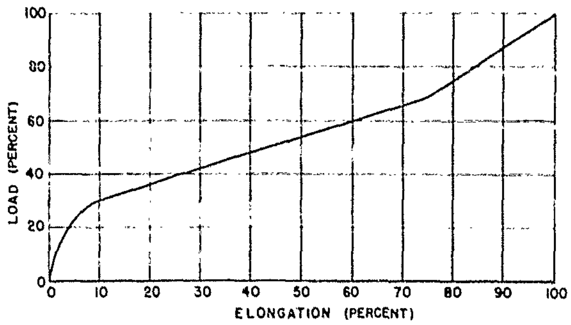


Figure 5 - Characteristics of elastic shock cord

opposing requirements led us to the selection of 75 percent elongation as the maximum operating limit of the shock cord, and 4 ft as the maximum carriage travel. The curve shape also determined the length of the wire rope tow cable, such that the test carriage strikes the stopping device with about 10 percent elongation left in the shock cord. This "tends to linearize" drop height with velocity and reduces undesirable bounce oscillations after striking the stopping device. The curve shape shown in Figure 5 is characteristic of all diameters of shock cord up to about 1 in. We selected a 5/8-in. diameter cord, a single strand of which produces a force of 330 lb at 100 percent elongation. The number of cords and their length was so selected that the necessary energy was available at 75 percent elongation. The resulting release acceleration was calculated and these parameters were then adjusted to give the most efficient design.

The stopping device for this drop tester is its most important and unique feature. It has been and remains the subject of considerable experimental research. Our experience to date will be outlined. However, new knowledge is being added almost daily.

The desired characteristic of a stopping device is its ability to absorb energy nonelastically and yet remain reproducible, predictable, and reuseable. Three general areas, shown in Table 2, occupied our attention.

Mechanical devices are limited because of frequency problems, unpredictable friction forces, and expense in first cost and maintenance. The deformation of materials either by direct crushing, shearing or buckling has generally been unsatisfactory for this application because of the large "spike" of acceleration produced on initial contact of the metal surfaces when the striking velocity is of the order of 80 fps. The success with deformation of nonmetallics has been varied; the limitations are generally due to temperature and humidity effects or nonreuseability.

Sand-filled rubber bags (Figure 6) have been our most successful stopping devices. These represent a compromise in the elastic and plastic absorption of energy. However, we have found them to be very reproducible, predictable and reuseable. Over 500 drops have been made on the type of bag shown in Figures 3 and 4 with no perceptible change in characteristics, or deterioration. The physics of energy absorption of this type of stopping device encompasses four types of phenomena:

1. Lateral expansion of the rubber bag
2. Compaction of the sand
3. Internal friction of the sand grains
4. Fracture of the sand grains

The parameters of the sand-filled rubber bag which control the types of deceleration pulse are as follows:

1. Stiffness. This parameter controls the peak g in the shock pattern. The rubber wall affects stiffness; this stiffness must be minimized, however, to reduce the elastic absorption of energy. Although the rubber durometer and composition are important in this factor, we have found that the addition of wire rope hoops around the bag as shown in Figure 6 is very effective in changing this parameter.

2. Shape. The parameter of shape controls the build-up time of the shock pattern. If the cone on the rubber bag is severely truncated, a very

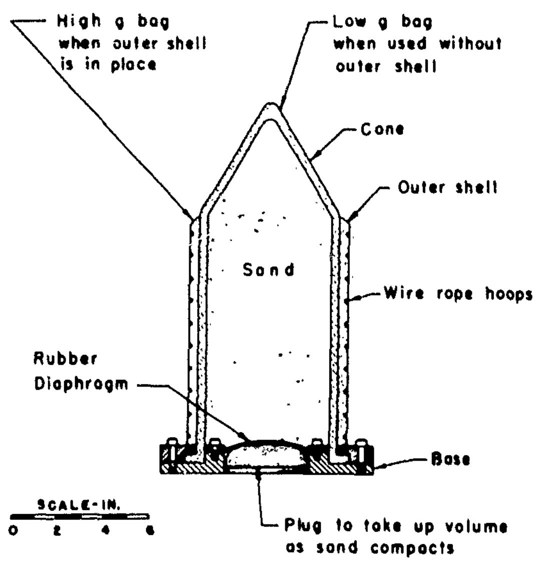


Figure 6 - Schematic of stopping device

sharp build-up time will be experienced. This time can be lengthened by having more and more of the cone participate in the initial deceleration of the carriage. Polyphase shock will result if a hemispherical-top bag, flat-top bag or short-cone bag is used. At velocities higher than 80 fps the cone must be longer to hold a constant build-up time and insure a smooth pulse.

TABLE 2
Limiting Characteristics of Stopping Devices Evaluated for Use on Mk 7 Mod 0 Drop Shock Tester

Classification	Item	Limiting Characteristic
Mechanical	1. Mechanical brake with leaf springs	Natural frequency and variable friction which cause shock pulse to be nonreproducible and extremely erratic; first cost and maintenance high
	2. Ramrod thru sand	Reproducibility poor, temperature and humidity effects on sand; sand maintenance problems
Deformation of Metals	3. Lead slugs	OK < 30 fps; "spike accelerations" > 30 fps
	4. Lead shells (Buckling)	"Spike accelerations," buckling produces ripples in pulse
	5. Thin walled .020 in. or less steel shells (Buckling)	Similar to lead but with sharper peaks, good energy absorber
Deformation of Nonmetallic Materials	6. Fiberglass packaging materials	Spike on end of pulse as cushion bottoms; reuseability poor
	7. Foam rubber	Spike on end of pulse as cushion bottoms; elastic absorption too great
	8. Cork cylinders	Good pulse shape but cork disintegrates above 40 fps
	9. Beeswax and microcrystalline wax	Good pulse shape; temperature-sensitive, reuseable only with remolding; may shatter at high velocities
	10. Sand-filled rubber cylinder, open top	Good pulse; sand scatters; some variation from temperature and humidity effects
	11. Steel shot-filled, rubber bag	Rectangular type pulse; fewer g's per unit of energy absorbed than sand; heavy; compaction of shot bad
	12. Sand-filled rubber bag, cone top	Good pulse; reproducible; reuseable; insensitive to temperature and humidity effects. Requires mechanical take-up to accommodate volume change of sand compaction and break-up
	13. Sand-filled rubber bag, cone top, wire rope hoops	Improved nonelastic energy absorption to give more g's increase in resistance with little increase in weight; number of wire rope hoops may be varied to give varying resistance

Note: The above results are dependent upon carriage weight and natural frequency.

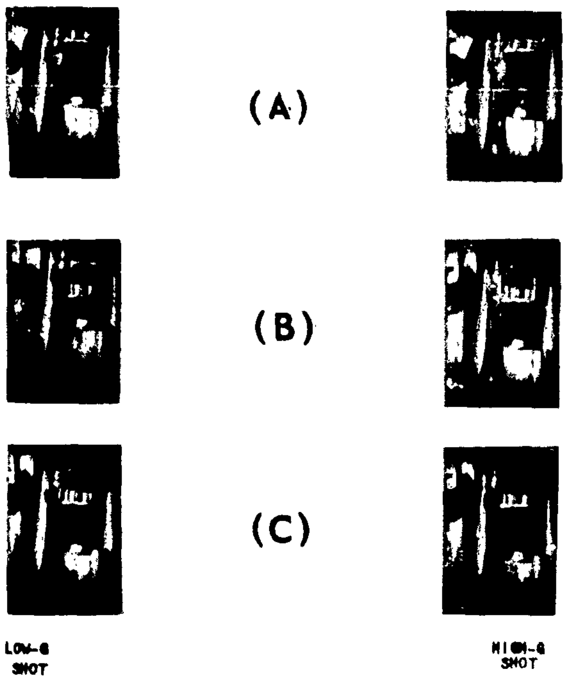


Figure 7 - Deflection characteristics of low-g bag on left and high-g bag on right: (a) before impact, (b) maximum deflection, and (c) final position

3. Bag height and diameter. These two factors predominantly affect the volume of sand and rubber which participate in the deceleration of the carriage. Variations in them, to a large extent, control the total energy which can be absorbed by the stopping device.

The above parameters can be varied in such a way that the shock pattern can be tailored to many varied requirements. Two stopping devices have been standardized for use in the Mk 7 Mod 0 Drop Shock Tester. They are identical in all parameters except that the high-g bag has a cylindrical sleeve, made of rubber and fitted with wire-rope hoops, which slips over the low-g bag (Figure 6). The deflection characteristics of the low- and high-g bags are shown in Figure 7. These pictures are frames from high-speed movies. The low-g shot was a 36-in. drop resulting in a velocity of about 75 fps and 165 g's. The high-g shot was at the same velocity but resulted in a much higher g value.

An important factor in attaining reproducible shock patterns with a drop tester is the foundation supporting the stopping device. Since a portable drop tester may be located on floors of varying rigidity so that a semipermanent, massive anvil may be impractical to use, the stopping device should be isolated from the floor foundation. In the Mk 7 Mod 0 Drop Shock Tester this is accomplished by seismically mounting the stopping device on a 500-lb anvil which is supported by two relatively soft springs. The natural period of this mass-spring system is about ten times as long as the shock pulse duration. Consequently, not only is the dynamic loading on the floor reduced but also the shock pattern becomes independent of the floor rigidity.

OPERATIONAL CHARACTERISTICS

The over-all acceleration time curve for this drop tester is shown in Figure 8 with the different phases of the machine operation indicated schematically. Note that the carriage experiences an acceleration building up to a peak of 50 g's in 3 to 5 ms as it travels downward a maximum of 4 ft. At this point it strikes the stopping device which starts the deceleration phase. The duration and peak values of this phase are controlled by the stopping device used—the shorter times and higher accelerations being associated with the high-g bag. The operational characteristics plotted as a function of drop height are shown in Figure 9. This chart can serve essentially as a calibration curve for the drop tester. For example, if it is desired to run a shock test with a peak acceleration of 500 g's, one enters this table at the 500-g point and by running up this line can determine that a 33-in. drop height will be required to produce this acceleration with the use of the high-g bag. By following horizontally along the 33-inch drop height line, one can determine that the velocity at this drop height would be about 65 fps as determined from the upper velocity scale, and similarly by reference to the release curve that the release acceleration would be on the order of 35 g's. These curves show the interrelationship between drop height, velocity, and

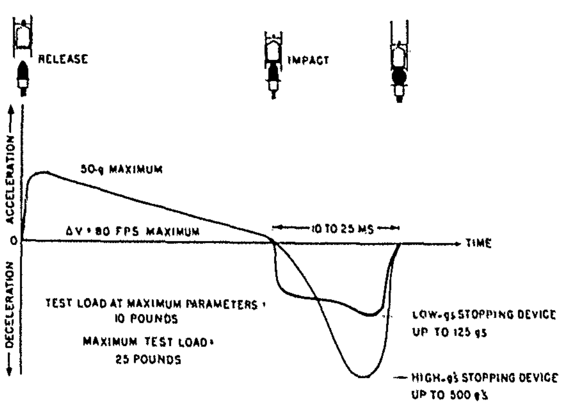


Figure 8 - Schematic acceleration-time curve

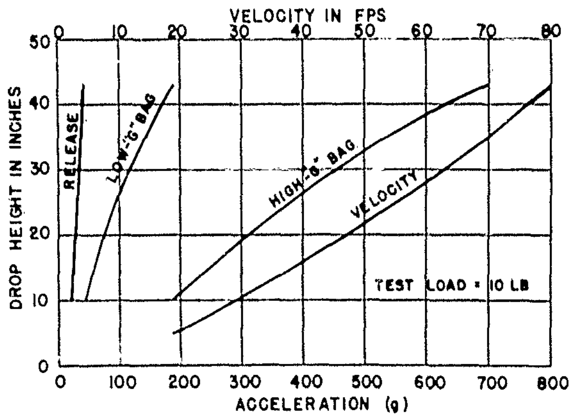


Figure 9 - Operational characteristics of drop shock tester Mk 7 Mod 0

acceleration which can be produced in the drop tester when the test load is 10 lb. Similar curves would have to be determined for test loads different from this value.

SUMMARY

The Mk 7 Mod 0 Drop Shock Tester is a low-cost, portable machine which is simple to operate but which is universal in application. It is intended to be used for making shock screening tests in quality control programs in ordnance manufacturers' plants. Its primary limitation is in the size and weight of specimen which it can accommodate. However, the design of this machine is such that it can be readily modified to accommodate larger and heavier specimens without serious compromise of the basic design principles.

It was the objective of the Naval Ordnance Laboratory to establish and evaluate the design of the subject machine. This has required about one man year of effort over the last two calendar years. The major contributions to this design were made by Mr. V. F. DeVost of the Shock Test and Design Section of our Laboratory. However, there are four other contributors to the patent disclosure. Although the patent rights on this machine are assigned to the Government, (Serial No. 345,272, Navy Case No. 14,587 filed 27 March 1953) a royalty-free license will be granted to any qualified contractor, company or agency to manufacture, use and/or sell this machine on request to the Naval Ordnance Laboratory, Silver Spring, Maryland. Likewise Ordnance drawings (BuOrd LD 290962 and all drawings listed thereon) completely disclosing the design and manufacture of the machine are available through this Laboratory.

DISCUSSION

M. Goldberg, BuOrd: What steps have been taken to authorize the use of this equipment in specifications?

New, NOL: The machine is disclosed by ordnance drawings of its design. Certain specifications which control some of the ordnance work at NOL, soon will require—or will suggest—that this machine be used for the shock tests.

L. A. Danse, General Motors: You mentioned the use of sand in the cushioning bags. Have you noticed the differences between the types of sand available; for instance, the difference in grains of the so-called Ottawa silica as against bank sand or ordinary shore sand?

New: No study has been made of the properties of sand for this application. The sand that has been used, however, has been subjected to considerable crushing.

Danse: If you check on sands you will find that some crush less than others.

S. Schwartz, NAES: How do you maintain calibration?

New: The shock characteristics of the machine can be reproduced very well. The shock cord has good elastic properties; the calibrations check after 300 shots. Very little change has been found in the elongation curve. However, in case a change occurs, it is a very simple operation to insert the dynamometer, shown in Figure 4, and obtain data for a new elongation curve.

Schwartz: How about the load?

New: The anvil doesn't change. The bags have withstood several hundred shocks. Photographs of consecutive shock pulses may be superposed to show how well the pulse is repeated.

Schwartz: How about the pickup?

New: A pickup is unnecessary because the shock pulse is repeated. For a given height of drop, the same pulse will be obtained, e.g., if 33-in. drops are made on the high-g bag, the 500-g pulse

shown in Figure 8 will be obtained, providing good practices are employed in loosening up the sand and getting it up inside the cone of the bag. A skilled operator is not required. All that is required is that the drop be made on to a bag that has been standardized. The calibration curve will give the acceleration and the duration of the pulse.

Schwartz: In Figure 5 the variables are percentage load and percentage elongation. In the first 10 percent of elongation the load rises very rapidly; then from 10 to 75 percent elongation, the load increases linearly and less rapidly. Consequently, the energy of elongation, proportional to the area under the curve, is less for the first 10 percent elongation than for, say, 10 to 20 percent or any further increase in elongation. Do you have any special reason for selecting the linear part of the curve, 10 to 75 percent elongation, for the operating range? What would be the result if the type and size of cord were changed?

New: You want the force bringing the carriage up to velocity to be as low as possible. On the other hand, you want to give the carriage as much energy as you can, so you try to get the maximum area under the curve. You get a good compromise of large energy against low initial accelerating force by selecting 75 percent elongation as the maximum operating limit. If you consider only the first part of the curve, you see it has little area for a relatively large force, which is not desirable. This is what gives "reverse" acceleration.

If the cord is changed (e.g., to a 1/2-in. size) the available energy will have to be redetermined.

Goldberg: Some of you may remember the pauper's version of the shock machine. It consisted of a simple wooden platform with pointed wooden posts projecting vertically downward. It was hoisted with block and tackle and then dropped onto a bed of sand, thus driving the pointed posts or spikes into the sand. This closely simulated the British shock machine. Furthermore it was reproducible, which we cannot say of the British machine. Two machines yield different results, depending on the way the equipment is mounted. I have recommended for years that we adopt some such machine that is reproducible in place of the British machine. Equipments could then be compared; this cannot be done at the present time. I think a machine of the type that I have recommended is cheaper to construct, is adequate and more reliable.

New: One of the purposes of presenting the 100-ft Drop Tester at this Symposium is to make it available to the maximum number of people for the minimum amount of effort on our part. We hope the people who have a use for it will work it into their problems and their specifications.

J. Steinman, Hughes Aircraft: Mr. New, now that you have this shock machine which is reproducible, I would like to know if there is some background for specifying the particular type of shock and the subsequent use of the machine?

New: The particular pulses were required for a certain problem that has been studied at NOL. However, in the course of experimentation the pulse was found to be subject to considerable variation as a result of changes made in certain parameters such as changing the type of bag. Now, for a different service environment it may be that the shape of the pulse should be changed. This can be done and standardization can be obtained on certain types of bags. The bag now in use gives the pulse experienced by a piece of ordnance equipment upon water entry. Some modification will be required for the bag to give a pulse that simulates the shock experienced by some other item of equipment under different conditions of shock.

C. E. Crede, Barry Corp: Mr. New mentioned that he put a steel plate on the bottom of the carriage to obtain additional rigidity. What is the lowest natural frequency of the carriage structure?

New: The lowest natural frequency is around 1200 cps.

Major E. J. Simmons, CSRDE: I would like clarification of the comment on the British machine.

Goldberg: The machine referred to is the British light-weight H. I. Machine, used to shock test light-weight equipment.

Simmons: It is my belief that the problems to be solved on the machines are not the same. For a 5-ft hammer drop the H. I. machine gives a much higher acceleration, about 4000 g's on the anvil plate for the back blow, about 2500 g's for the side blow, and about 2000 g's for the top blow as compared to a maximum value of 500 g's for the Drop Tester. Also, the time interval to maximum acceleration is less than 1 ms for the light-weight H. I. Machine, whereas the impact of the other machine is of longer duration and may be classified as a medium impact or the so-called impact for variable tank variations.

C. E. Taylor, Portsmouth Naval Shipyard: I feel that more should be said about the British H. I. shock machine. On this machine items up to 250 lb in weight can be tested, whereas on the 100-ft Drop Tester the weight limit of items to be tested is 20 lb. Also, I believe I am right in saying that the range of shock is entirely different. The H. I. shock test is more severe than the shocks normally received by equipments during the period of installation on board ship, whereas the drop test is much less severe.

New: The speaker did not intend to convey the impression that the 100-ft Drop Tester replaces the H. I. machine in any respect.

* * *

PROPOSED TESTS FOR ESCAPE FROM VERY HIGH VELOCITY AIRCRAFT

Lt. Col. J. P. Stapp, USAF (MC) and 1st Lt. H. P. Nielsen, USAF

A rocket-powered sled on a 3500-ft track decelerated by a water scoop will be used to reproduce time-deceleration curves expected in escape from aircraft at velocities up to 1800 mph at 40,000 ft. Subjects will be instrumented to determine physical forces and biological effects.

As we see it, the problem facing aircraft designers is vastly different from that facing the people involved with the human factors of high-speed and high-altitude flight. Aircraft designers have the opportunity to scrap old models and designs that do not meet known design specifications, but those interested in human factors are not quite sure to what specifications their standard operational model was built and once they find out, they are not able to scrap what they have for a new version.

A crew member escaping from an aircraft at high speed and high altitude is exposed to numerous hazards up to the time of parachute deployment at, for example, 15,000 ft. These are:

(1) low temperature

(2) low atmospheric pressure,

(3) tumbling and spinning

(4) wind-blast, and

(5) deceleration.

Clothing with 2 to 3 equivalent clo insulation will adequately protect a man against the low-atmospheric temperatures during a free fall from 100,000 ft to 15,000 ft (Reference 1). The time for this long drop has been calculated to be approximately 3-1/2 min (Reference 2).

The Air Force T-1 altitude suit, which is currently undergoing standardization, has been tested to 106,000 ft, the limit of the Aero Medical Laboratory's altitude chamber. Three subjects have worn the suit for periods from 2 to 7 min of this altitude breathing oxygen at a pressure of 150 mm of Hg (Reference 3). The new pressure helmet will withstand a Mach 1 wind-blast at sea level.

Tumbling and spinning may present problems which are difficult to resolve. Bodily rotation of an escaping air crew member may be divided into two main types.

(1) Tumbling - which we define to be the head-over -heels rotation in a vertical plane during the ejection process, and

(2) Spinning - which we term to be the rotation of the body in a horizontal plane and about a vertical axis.

Figure 1 is a composite photograph of an actual dummy test ejection. The photographs were taken at 128 frames per sec and every sixth frame was used in making this photograph. This ejected dummy and seat combination rotated at more than 180 rpm from the time of ejection until it passed over the tail. At low altitudes tumbling is fairly well damped out in 3 to 5 sec but would persist for longer periods at high altitudes (Reference 4).

Captains Weiss and Edelberg of the Aero Medical Laboratory, by extrapolation from their animal and human experiments on the 8-ft horizontal spin table, have established 3 to 10 sec as the time limit of consciousness while rotating at 180 rpm with the center of rotation through the iliac crest (Reference 5).

The British in one out of a series of 40 dummy free falls measured peak rotational speeds of

Figure 1 - Dummy test ejection

240 rpm while the dummy was spinning in a horizontal plane about a vertical axis (Reference 6).

At 240 rpm and with a center of rotation through the iliac crest, blood pressures at the eye level would reach values roughly equivalent to 18 negative g's.

Since the Air Force's recommended procedure following ejection calls for separation from the seat after only a few seconds (Reference 7), the crew man will be free of the seat for almost all of the descent. Therefore, if he encounters spinning and is unable to counteract it, he will be subject to the spinning for a long period of time. A test program is being initiated this fall to determine the rates and centers of rotation that may be experienced by a man making a delayed free fall. This program may indicate that during descent from high altitudes stabilization of the man, either alone or in a seat or capsule, must be accomplished to prevent the man from reaching rotational speeds beyond his physiological tolerance.

Recent experimental evidence tends to support the belief that wind-blast is not the terror we have been led to expect.

In February 1952, two 10-lb monkeys were wind-blasted at 790 and 820 mph respectively, at sea-level conditions (Reference 8). The animals not only survived, but 24 hours after exposure showed no effects of their experience with the exception of the second monkey who suffered a pair of black eyes when the eyelids were fluttered very rapidly by an air stream entering the protective helmet through a small hole. Figure 2 was taken during the height of the 820-mph wind blast and despite the way the monkey's body was contorted by the force of the blast, the animal suffered only from mild shock and the black eyes mentioned.

On 9 and 10 April of this year two 100-lb chimpanzees were subjects for wind-blast tests on the 10,000-ft track at Edwards AFB. Figure 3 shows the prerun test configuration.

Both animals received a wind-blast of approximately four seconds duration above 600 mph with a peak speed of approximately 800 mph.

There is a point of special interest in these chimpanzee tests: head and face protection failed early in the run in both cases and yet the animals suffered no significant injuries to internal organs or face. The second animal did suffer eyes swollen shut and rather severe lip abrasions but this damage was due to being beaten in the face by the zippers on the neck curtain of the specially designed protective helmet. X-ray photographs showed no lung damage in either animal. Figure 4 shows subject No. 1 after the run.

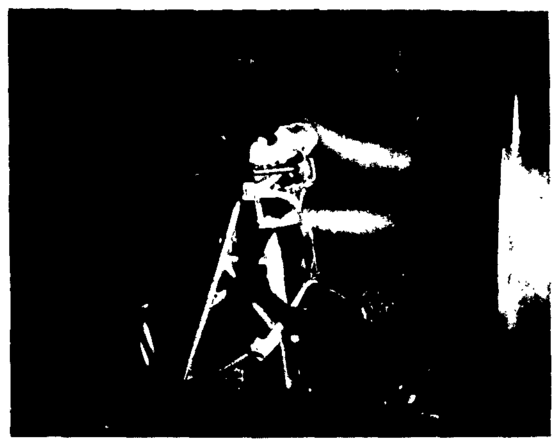


Figure 2 - Test configuration of monkey subject during the height of a 820-mph wind blast

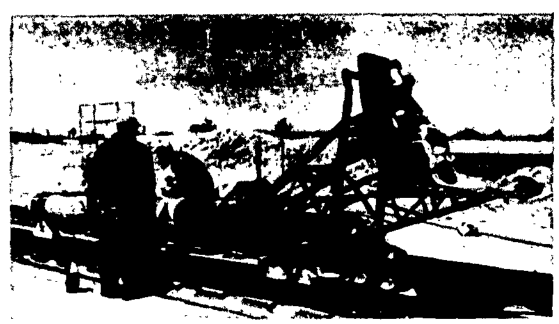


Figure 3 - Prerun test configuration for wind-blast test on chimpanzee

Figure 4 - Chimpanzee after being subjected to an 800-mph wind-blast test

It seems that the first animal lost its face protection either at or right after peak speed since the helmet was found at station 1800 ft while the peak speed was reached at station 1200 ft. Note that there is no visible facial damage but that there was considerable fluttering of the clothing as indicated by shredding at the shoulders and knees. It has been found necessary to restrain the limbs to prevent them from flailing about and being injured. You will note that in both the small monkey and the chimpanzee tests this precaution was taken. In May 1952, Andre Allemand, a French test pilot suffered permanent injuries to both hip joints due to wind-blast forcing his legs apart following a test ejection from a Gloster Meteor (Reference 10).

It would seem logical that ejection seats in future high-performance aircraft should have better provisions to restrain the legs and arms to prevent injury to them during both the ejection and wind-blast.

If we expect men to fly high-performance aircraft safely and effectively, human factors research on man's capabilities must keep ahead of aircraft design. In order to permit a safe escape from a crippled aircraft, we must know man's specifications. To help achieve this end, a program has been initiated to explore the limits of human tolerance during very-high-speed escape relative to the factors of deceleration, tumbling, and wind-blast, either singly or in combination, by means of a high-speed sled, track, and appropriate braking system. Although wind-blast and tumbling respectively, can be studied readily with conventional type equipment, the deceleration following ejection from high-speed aircraft can best be reproduced experimentally with a high-speed sled decelerated by water brakes.

Dr. Scherberg and Mr. Ferguson of the Air Force Flight Research Laboratory presented at the 19th Shock and Vibration Symposium in September 1952, a curve showing the deceleration pattern to be expected by a man-seat combination ejected from aircraft (Reference 11) which are either flying now or will be operational in the near future. The solid curves in Figure 5 show these data and the dotted curves show the deceleration-time history of two actual human ejections (Reference 12)

The reproduction of the deceleration pattern forecast by Scherberg and Ferguson is not possible by means of the mechanical brakes used with the track at Edwards AFB by Col. Stapp in his early investigations, but is possible with water brakes.

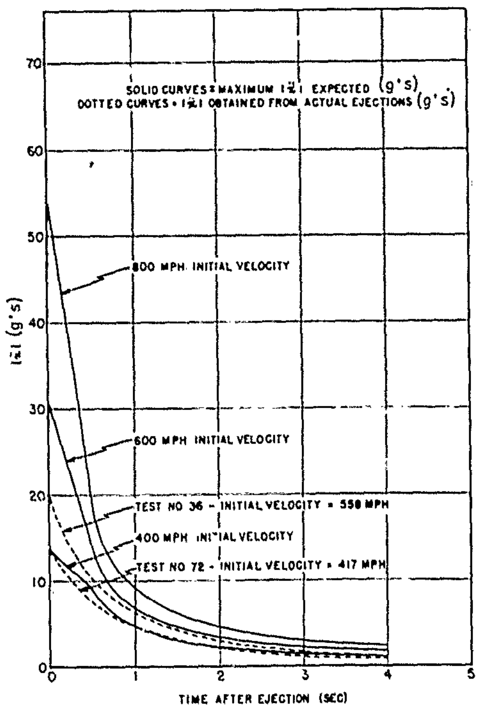


Figure 5 - Deceleration patterns of man-seat combination ejected from aircraft

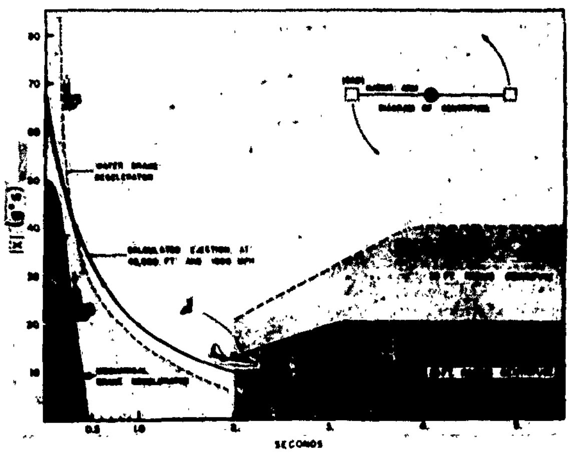


Figure 6 - Relative performance characteristics of various deceleration devices as compared to a representative acceleration-time history for a man-seat combination ejected from a high-speed aircraft

Figure 6 shows the relative performance characteristics of various deceleration devices as compared to a representative acceleration-time history for a man-seat combination ejected from a high-speed aircraft.

The water braking system consists of a water scoop on the sled and a water trough between the track rails. As the sled and scoop proceed down the water trough, the scoop picks up the water, and by turning it approximately 180°, causes a decelerative force to be applied to the moving sled.

The decelerative force is a function of the mass rate of water pick-up while the "jolt" or rate of onset of deceleration is dependent upon how the mass rate of water pickup varies with time. Both of these factors are changeable between runs.

The trough, which is 2000 ft long, can be divided by dams into sections 10 ft long with the depth of water varied from one section to another.

By this braking arrangement, Northrup Aircraft Incorporated, designers and builders of the equipment, will be able to vary the peak deceleration from 25 to 2000g's and the rate of onset from 100 to 5000 g's/sec. Expected performance is shown in Figure 7.

The vehicle to carry the test subjects will be pushed by a propulsion unit designed to utilize either solid propellant rockets or a liquid fuel rocket engine. Maximum velocity of the test vehicle into the braking area will be approximately 750 mph.

The test vehicle, Figure 8, is designed to permit the study of deceleration of animal and human subjects in the forward facing position to simulate emergence from the cockpit. Exposure in the rearward facing and in headfirst or feetfirst positions, will permit evaluation of deceleration in corresponding body positions following exit from the aircraft.

A limited number of human decelerations will be performed after animal and instrumented dummy runs have proven the feasibility of human tests.

To study wind-blast and tumbling, a seat mounted on gimbals attached to the sled will be exposed to the air blast by the removal of a wind-screen just prior to the time when the vehicle enters the braking area. The magnitude of the wind-blast realizable is dependent solely upon the speed of the sled before and after entry into the water brakes.

Tumbling of the seat will be initiated by an ejection seat gun as the sled enters the braking area. In this manner the subject will be successively subjected to wind-blast, simulating the blast encountered by removal of an aircraft canopy; then to wind-blast, tumbling, and deceleration simulating conditions during and after ejection.

With a 100-lb subject in the rotating seat, rotational speeds up to 180 rpm may be induced

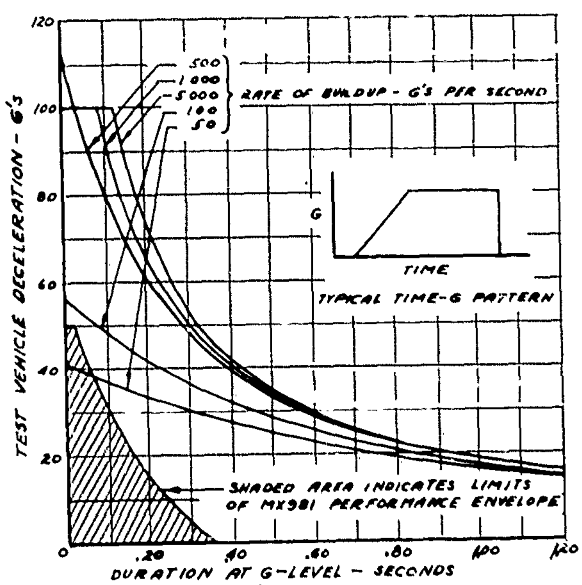


Figure 7 - WADC Aeromedical Laboratory decelerator performance (maximum vehicle velocity = 750 fps)

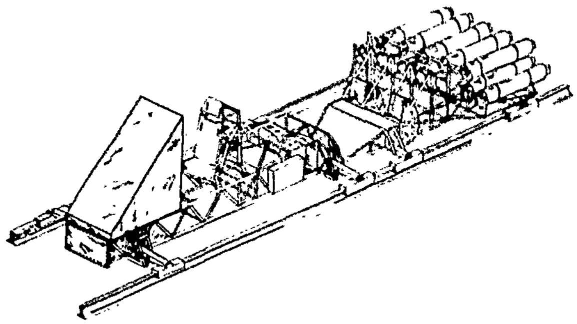


Figure 8 - Linear decelerator vehicle, Aero-medical Laboratory, WADC

while in a 50-linear-g field. With a 180-lb subject, tumbling at 180 rpm in a 25-linear-g field may be accomplished. In both instances a brake will arrest the rotation within 2 sec.

It is anticipated that the following program will provide a basis for evaluating ejection seat limits, capsule requirements, and ultimate limits of escape using ejection seat or capsule:

1. Runs to check equipment performance
2. Decelerations of instrumented dummies and chimpanzees to provide human configuration and viability data respectively.
3. Combined tests on chimpanzees followed by a limited number of human decelerations and combined tests.

This information will be the subject material for future reports.

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3. Verbal Coordination - Major Kennedy, Respiration Section, Aero Medical Laboratory, WADC, Wright-Patterson Air Force Base, Ohio

4. Verbal Coordination - Mr. K. Hecht, Aircraft Laboratory, WADC, Wright-Patterson Air Force Base, Ohio

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FLIGHT VIBRATION CHARACTERISTICS OF F-86-A5 AIRPLANE WITH MACHINE GUNS FIRING

D. C. Kennard and V. C. McIntosh, WADC

The flight characteristics of the F-86-A5 airplane with machine guns firing, as determined during a series of 34 test flights, are shown in graphic form. The data were obtained by use of a special remote control magnetic tape recorder, and harmonically analyzed by means of an automatic wave analyzer system. Recordings were made for 4-sec durations during which the machine guns were fired in bursts lasting from 1.5 to 2 sec. The levels of vibration for different structural components of the aircraft are given for various frequency ranges. Also, specifications for a complete magnetic tape system are summarized.

INTRODUCTION

The Equipment Laboratory, WADC, has conducted an extensive study of the flight vibration characteristics, from an equipment standpoint, of the F-86-A5 airplane (Figure 1). More than 34 flights have been made with guns firing to determine the attendant vibration effects on equipment and equipment locations. Data from these flights were analyzed and assimilated almost as rapidly as the flights were run off, a feat hitherto impossible with available instrumentation and techniques.

INSTRUMENTATION

MB Type-124 velocity-type pickups were used to sense aircraft structural vibration at the following locations: left wing tip, left wing frame near wing flap actuator, and aft section (Figure 2). Of particular interest was the main 400-cycle inverter located in the nose section. Pickups attached to the rigidly mounted inverter frame indicated essential structural vibration, as well as inverter vibration, in this region. Pickups were also located in the inverter control box, and the vibratory response of inverters with and without shock mounting of the inverter assembly was obtained. A Gulton Model A-104 accelerometer was used to measure the vibration characteristics of the elastically suspended voltage regulator within the inverter control box. Pickups were also located on the instrument panel, wing

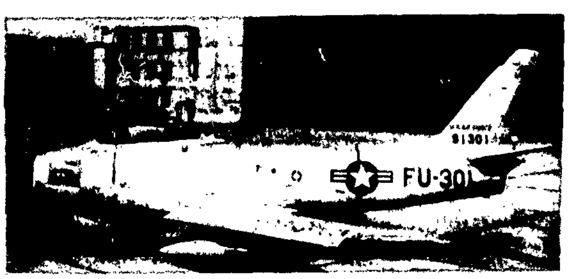

Figure 1 - F-86-A5 airplane

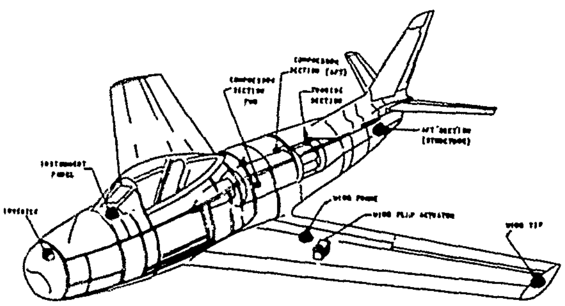

Figure 2 - Schematic diagram of vibration pickup locations on F-86-A7 airplane

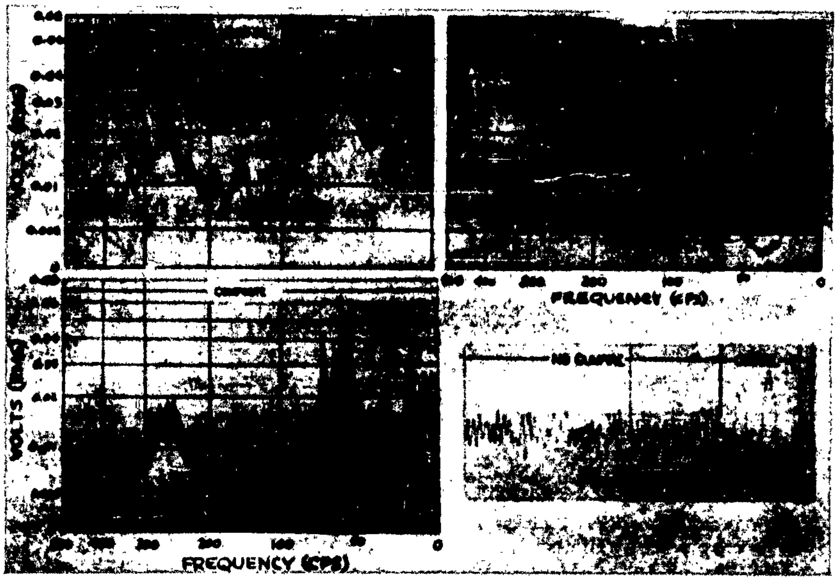

Figure 3 - Composite and individual analyses of flight vibration characteristics of F-86-A5 with and without gunfire. Locations, aft section; direction, lateral; pickup, MB Type-124.

flap actuator, and on the turbine and compressor sections of the engine.

All vibration recordings were made by the pilot, using a special remotely controlled magnetic-tape recorder. Magnetic-tape recordings subsequently were analyzed harmonically in the laboratory by means of an automatic playback and wave analyzer system. The summary of data which follows is presented as an example of the type of information that can be obtained by means of this magnetic-tape-recording, playback, and analyzing system which provides a new and powerful tool in vibration-measuring technique.

RESULTS

Vibration data were obtained from tape recordings of 4-sec duration during which the machine guns were fired in a burst of 1.5- to 2-sec duration for each recording. Hence, when played back in a continuous loop, each recording indicated a high and low level of vibration. Figure 3 shows a sample of the composite, high- and low-level, harmonic analysis plotted on a Brown strip recorder. The gun-firing portion of the tape recording was then cut out and played back as a continuous loop. The resulting harmonic analysis shows the same contour as that of the high-level composite analysis. The low-level portion of the tape also was played back, and duplicated the low-level contour. A portion of the actual velocity trace, including the high- and low-level conditions of vibration, also is shown in the figure.

At the wing tip, maximum vibratory excursions were measured in the vicinity of 12 cps, amounting to 0.220-, 0.012-, and 0.038-in. double amplitudes in the vertical, lateral, and fore and aft directions, respectively (Figure 4a). Double amplitudes of 0.005 to 0.008 in. were measured in the 50- to 75-cps range in all directions. Vibration frequencies corresponding to the rotor speed of the engine were most discernible in the lateral direction, amounting to a maximum of 0.003-in. double amplitude. Presence of second-order engine vibration also was noted in the lateral direction (0.0002-in. double amplitude at 250 cps). Double amplitudes of 0.0002 in. were noted at frequencies as high as 310 cps in the lateral and in the fore and aft directions.

Vibration at the wing frame occurred generally at the same frequencies noted at the wing tip,

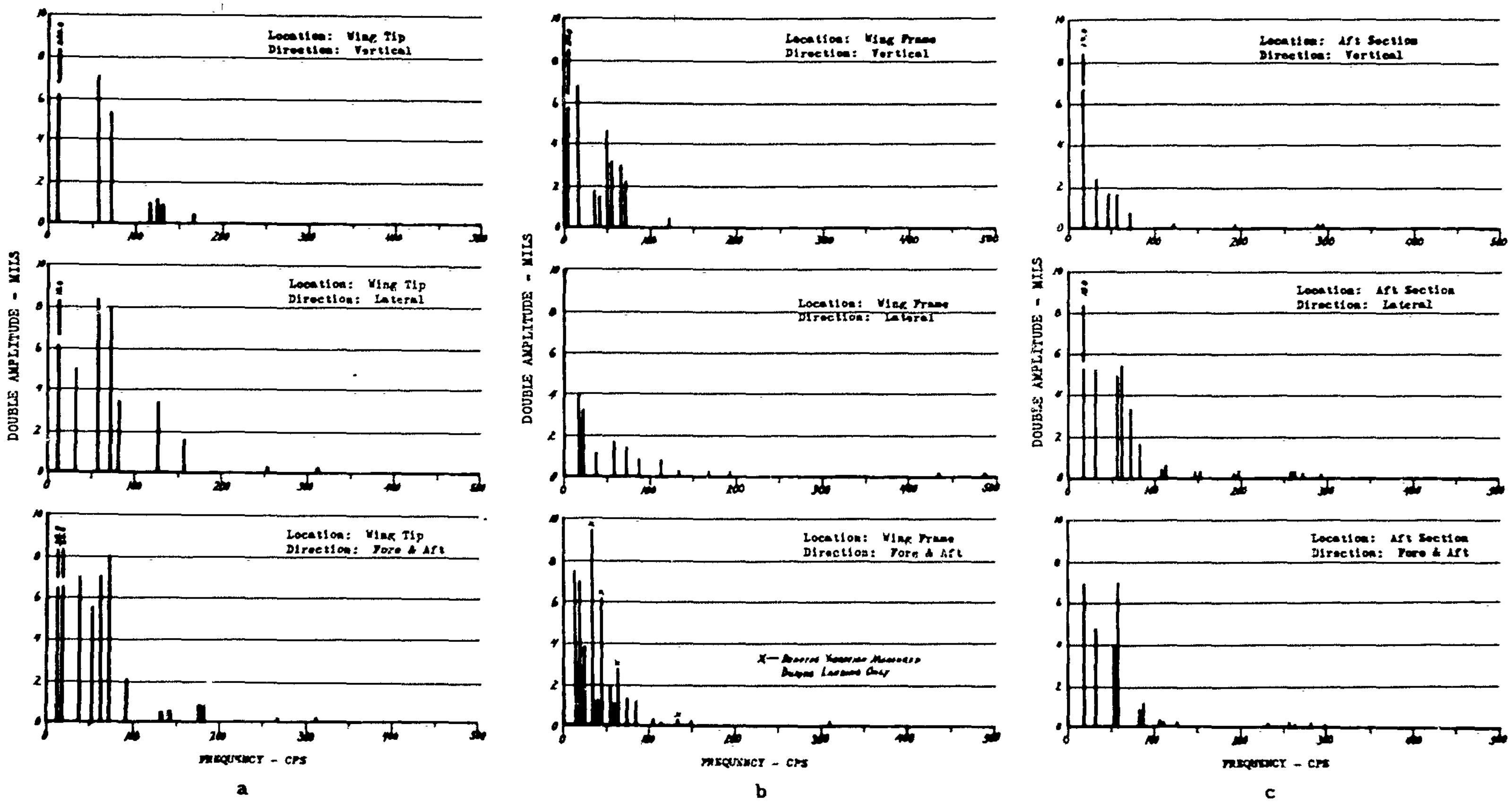

Figure 4 - Flight vibration characteristics of F-86-A7 Airplane with gun fire

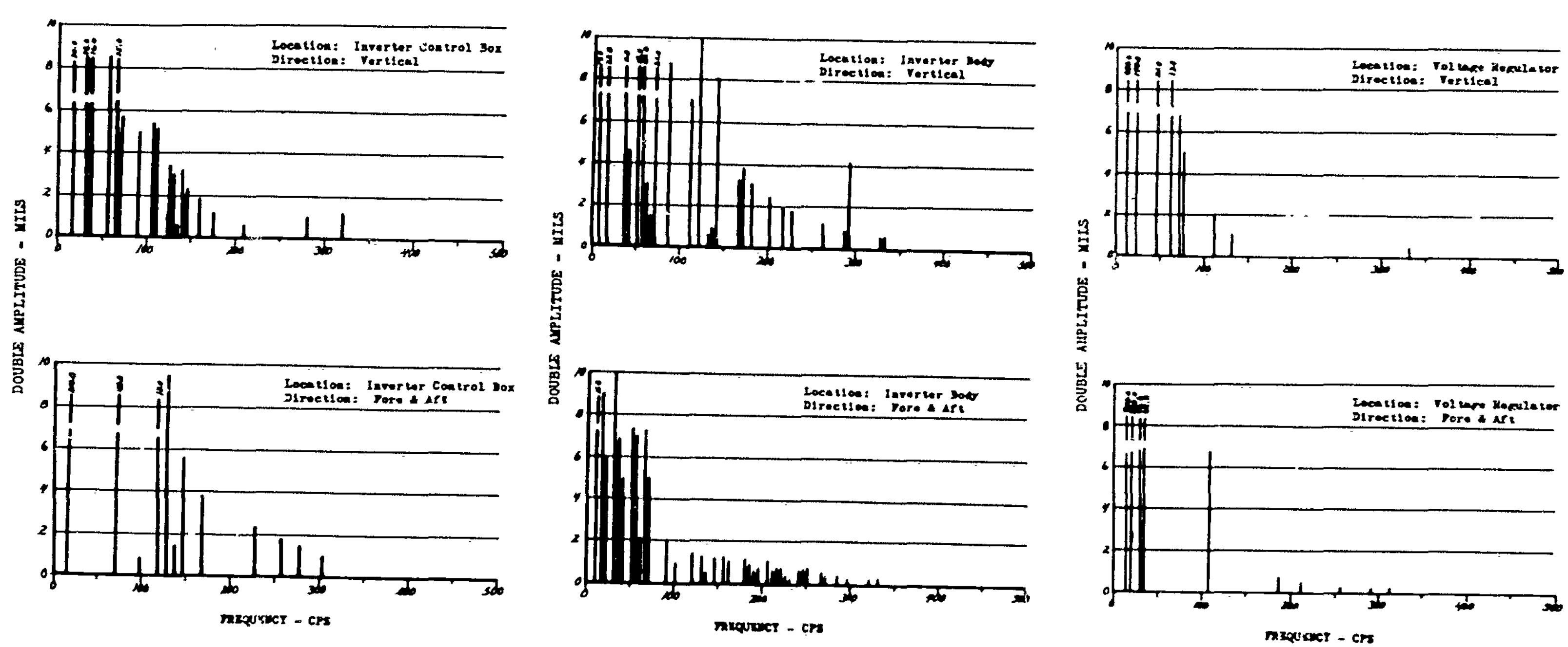

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Figure 4 (Cont'd) - Flight vibration characteristics of F-86-A7 airplane with gun fire

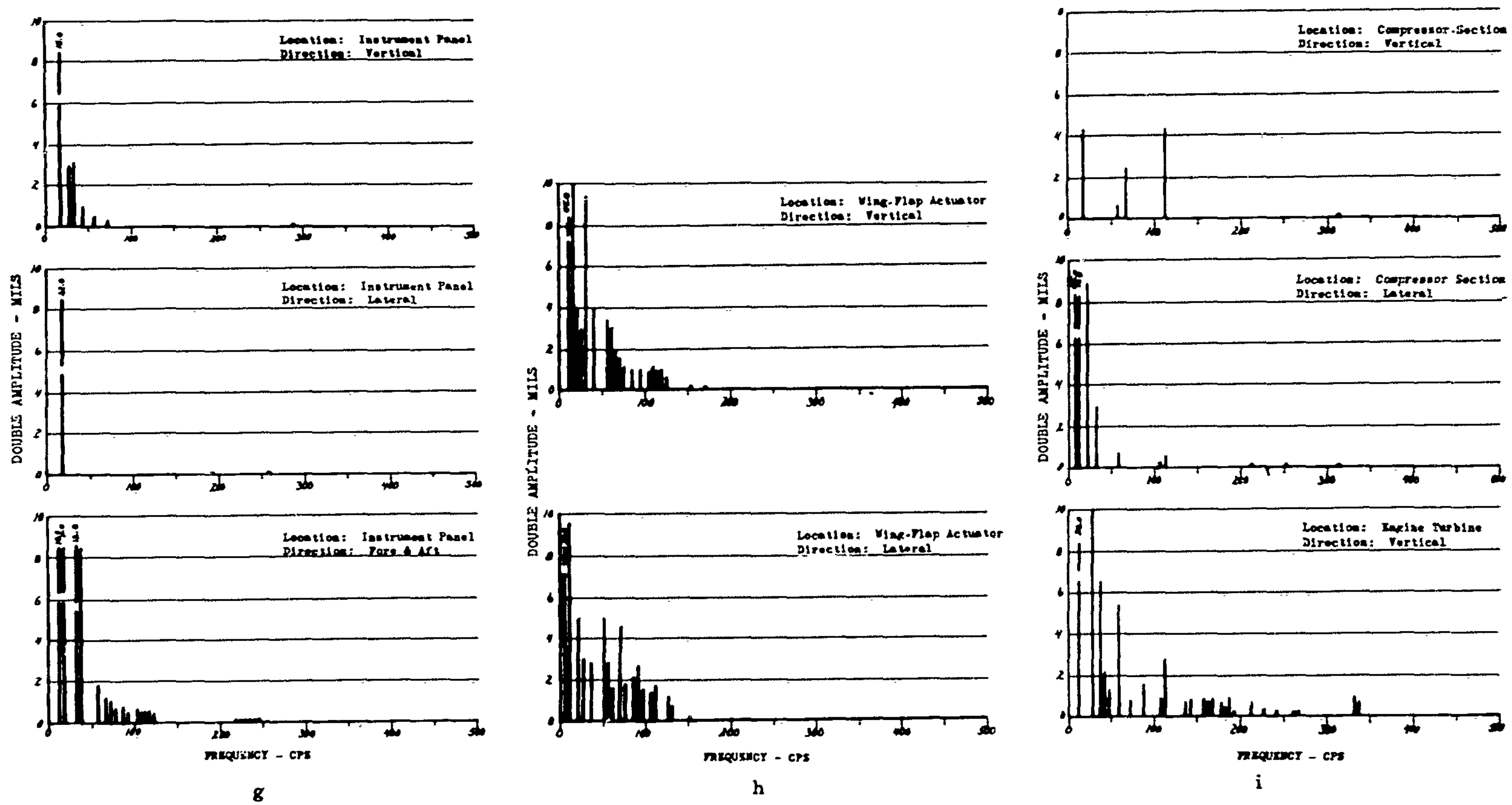

Figure 4 (Cont'd) - Flight vibration characteristics of F-86-A7 airplane with gun fire

but with reduced amplitudes (Figure 4b). Landing transients caused a maximum double amplitude of 0.009 in. in the fore and aft direction at a frequency of 30 cps.

In the aft section of the airplane the maximum excursion of 0.018- to 0.019-in. double amplitude occurred at 20 cps in the vertical and lateral directions (Figure 4c). In the frequency range of 55 to 60 cps a peak response of 0.005- to 0.007-in. double amplitude was measured in the lateral and in the fore and aft directions. Engine rotor frequencies amounted to less than 0.001-in. double amplitude. Evidence of frequencies as high as 300 cps was noted in all directions, the double amplitude being in the order of 0.0002 in.

In the frequency range of 10 to 70 cps the vertical vibration of the shock-mounted inverter body generally was less than that of the control box. In the control box (Figure 4d) the double amplitude varied from 0.008 to 0.038 in., whereas on the inverter body (Figure 4e) it varied from 0.003 to 0.022 in. Fore and aft vibration of the inverter body in the same frequency range varied from 0.002- to 0.011-in. double amplitude, whereas, in the same direction, there were peaks at 20 cps (0.021-in. double amplitude) and at 70 cps (0.040-in. double amplitude) for the control box. In the higher frequency range the vertical double amplitudes were generally greater on the inverter body than in the control box, although the inverse was true for fore and aft vibration. Frequencies as high as 330 cps were detected on the inverter—both on the inverter body and in the control box—the maximum double amplitude being 0.001 in.

The elastically mounted voltage-regulator assembly within the inverter control box vibrated with double amplitudes as high as 0.43 in. at 10 cps vertically and 0.90 in. fore and aft (Figure 4f). In the higher frequency range the most outstanding peak occurred at 110 cps, with 0.007-in. double amplitude in the fore and aft direction. Again traces of vibration as high as 330 cps were found.

Significant vibration of the instrument panel in all directions was limited to the frequency range of 10 to 40 cps, the double amplitude varying from 0.003 to 0.022 in. (Figure 4g). Presence of minor components of vibration was noted at frequencies as high as 300 cps in the vertical direction.

The wing-flap actuator responded both vertically and fore and aft, the double amplitude varying from 0.003 to 0.045 in. in the 5- to 40-cps frequency range (Figure 4h). From 50 cps the double amplitude in both directions decreased with increasing frequency from 0.003 and 0.005 in. to 0.0002 in. at 150 cps.

Engine double amplitudes were maximum on the compressor section, amounting laterally to 0.065 in. at 10 cps (Figure 4i). The maximum first engine order response was detected vertically at the compressor section, the double amplitude being 0.004 in. A double amplitude of 0.001 in. was measured vertically on the turbine section at 330 cps, but there was no measurable fore and aft vibration at this location. Tracers of second and third engine order were noted on the compressor section. As a matter of fact, presence of second and third engine order was indicated at all pickup locations except those on the wing frame, which showed nothing higher than first order.

These data are summarized from a total of 34 flights of approximately one-hour duration each. Only three flights were abortive, and these were due to checkout errors in instrumentation. Twenty-nine flights took place in the three winter months of January, February, and March. The guns were fired approximately 18 sec during each flight, and an average of ten recordings were made for each flight. Each recording produced vibration data simultaneously for 12 pickups. Only two engineers, working part time, were required to manage these flights and perform the subsequent analyses. It is quite apparent that the extent of this operation would have been considerably curtailed if conventional instrumentation had been used.

DESCRIPTION OF MAGNETIC TAPE EQUIPMENT

The magnetic–tape–recording, playback, and analyzing system which facilitated this work was manufactured to Air Force requirements by the Davies Laboratories, Inc. Its essential technical features are described as follows:

The system consists of a 14-channel magnetic-tape recorder, a 14-channel playback, and two automatic-recording wave analyzers (Figure 5).

Figure 6 shows the recorder installed in the F-86 airplane. It operates from a 22- to 28-volt dc supply and may be controlled remotely. The outputs of 12 velocity pickups and one engine tachometer can be accommodated simultaneously. By using frequency modulation, pickup voltages from 0.01 to 1.0 volt rms can be recorded through the frequency range of 3 to 2000 cps. The frequency-modulated carrier of 10-kc

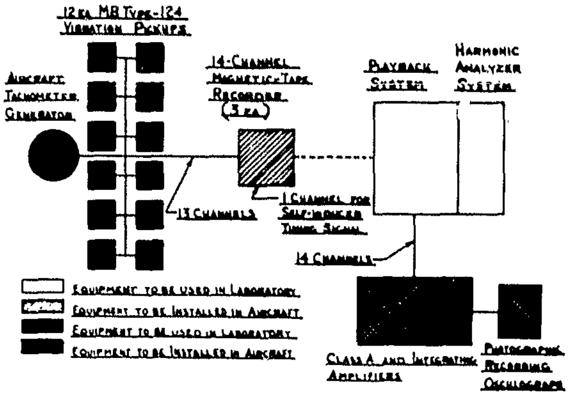

Figure 5 - Block diagram of vibration-measuring equipment

center frequency is applied through the recording head to a magnetic tape 1-3/4 in. wide. A crystal-controlled 25-kc oscillator provides a reference frequency which is recorded on the 14th channel. A 500-cycle oscillator is incorporated for channel calibration. The remote control indicates the amount of tape remaining and the calibration voltage. Lengths of records are timed automatically and may be set from approximately 1.5 to 5 sec. The record button bypasses automatic recording, so that longer records can be made. Tape speed is approximately 30 in./sec. Supply and takeup reels each hold approximately 350 ft of tape. Recording time per reel is approximately 2-1/2 min. A heating blanket is furnished with the recorder for operation at extremely low temperatures.

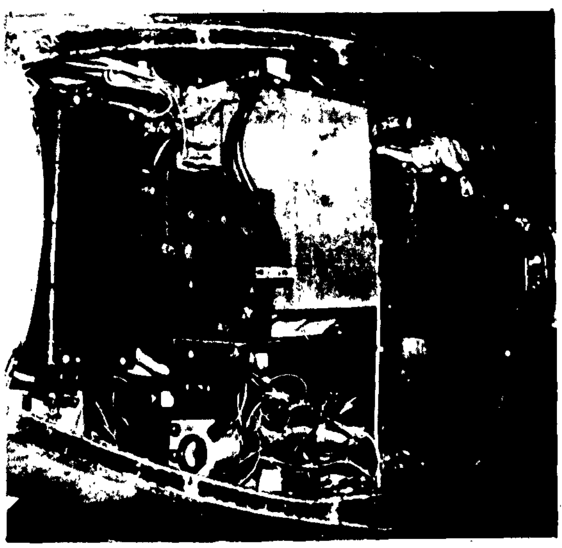

Figure 6 - Installation of magnetic-tape measuring equipment

The playback unit (Figure 7) demodulates and reproduces the original recorded signals. The average playback tape speed is servo controlled by the recorded 25-kc reference frequency. By mixing in opposite phase the discriminated output of the reference channel with each of the outputs of the signal channels, extraneous voltages due to residual tape speed fluctuations are eliminated. The playback contains a 500-cycle oscillator for channel calibration, and a frequency modulator similar to those in the recorder for adjusting channel discriminators and

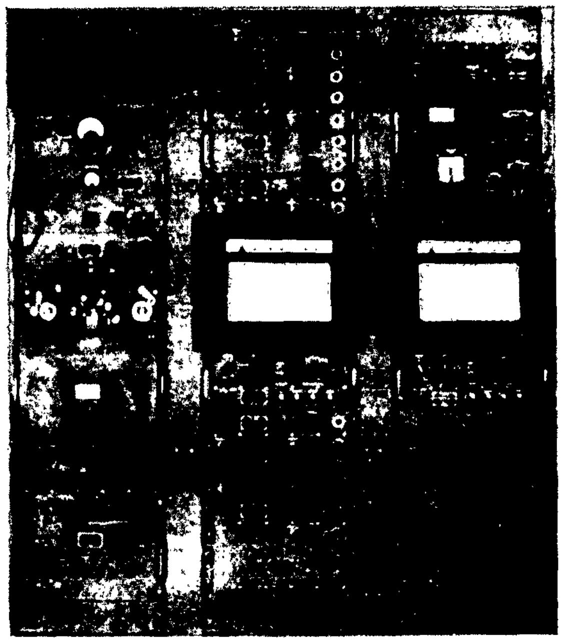

Figure 7 - Playback and wave analyzer racks

compensation circuits. The playback tape transport is capable of handling 350-ft reels of tape, or the tape may be spliced and played back in continuous loops from 4 to 75 ft in length. The played-back recordings may be recorded on conventional oscillographic equipment or processed by the automatic-recording wave analyzers.

Two automatic wave analyzers sweep the frequency range of two playback channels simultaneously and record the harmonic voltage components on two Brown strip-chart recorders. The paper speed of the strip recorders is synchronized with the analyzer frequency sweep, so that the record is a plot of harmonic voltage versus frequency. During automatic operation the strip-recorder paper speed is decreased relative to the speed of sweep at 100, 300, and 500 cps,

thereby condensing the frequency scale. Successive analyses of each pair of channels take place automatically up to a maximum frequency of 100, 300, 500, or 2000 cps, as desired. The strip-chart voltage scale is approximately logarithmic over the range 0.01-1.0 volt on a paper width of 8 in. A period of approximately 30 min is required to sweep each pair of channels up to 2000 cps, and 10 min is required up to 500 cps. The wave analyzers have two filter bandwidth ranges: $\frac{1}{3}$ - 8 cps and 10 - 45 cps. The broad range must be used for analyzing frequencies above 500 cps, because tape speed fluctuations otherwise carry the recorded frequency outside the narrow bandwidth. Frequency is indicated on the strip record by a separate marking pen.

SPECIFICATIONS FOR MAGNETIC-TAPE EQUIPMENT

Specifications for the complete magnetic-tape system are summarized as follows:

Recorder
Dimensions - 10 x 20 x 11 in.
Weight - 60 lb
Power requirements - 22-28 volts dc at 8 amp
Tape length per reel - 350 ft
Tape width - 1-3/4 in.
Tape speed - 30 in./sec
Input voltage, signal channels - 0.01 to 1 volt rms
Input voltage, tachometer channel - 0.48 to 48 volts rms
Calibrating voltage - 0.1 volt at 500 cps
Operating temperature range - +160 to -89°F (heating blanket used for temperatures below 0°F)
Relative humidity - near 100% without permanent damage
Vibration - Operates normally under vibration requirements of USAF Specification No. 41065-B, Method 61
Acceleration - Operates normally under a downward acceleration of 7 g's
Altitude - Operates normally at altitudes up to 50,000 ft

Playback and analyzer
Power requirements - 115 volts, 60 cycle, 2800 watts
Dimensions - 77 x 67 x 18 in.

Over-all performance
Voltage accuracy - ±5%
Frequency accuracy - 0.5% above 20 cps
Frequency response - Essentially flat from 3 to 1600 cps
Response off less than 3 db at 2000 cps
Rms noise level - less than 0.02 volt

CONCLUSION

With the aid of this new tool for obtaining flight vibration data, the Equipment Laboratory, WADC is looking forward to amassing a wealth of information on vibration characteristics of aircraft equipment with a minimum expenditure of techical manpower. It is anticipated that the knowledge thus gained will contribute materially towards increasing the effectiveness of our Air Force.

DISCUSSION

E. L. Eagle, Glenn L. Martin Co.: I noticed you gathered some high-frequency data. I was wondering if you compiled the information on the existence of high-frequency vibration while in normal flight.

McIntosh: I think, as a matter of fact, that most of our high-frequency data did not come from gunfire at all; that most gunfire frequencies were not above 200 cps.

H. K. Cheney, Consolidated Vultee: Where is that equipment operating now?

McIntosh: It is operating in the Equipment Laboratory at Wright Aeronautical Development Center.

O. Biamonte, Evans Signal Laboratory: Do you know what the frequency limitation is of the pickup that you use? Why do you use a velocity pickup rather than an accelerometer which could measure acceleration directly?

McIntosh: This equipment was designed for velocity-type pickups, which give a voltage output sufficient for the system at low frequencies, as well as at high frequencies. We did use one accelerometer with the equipment, and it worked out well. We measured low frequency vibrations on the inverter with it.

Biamonte: I was surprised not to see any of the higher frequency components. It seems to me that you showed frequency components only as high as 300 to 400 cycles, and previous reports indicated that frequency components were obtained as high as 600 cycles. I wonder what happened to the high-frequency components.

McIntosh: I will go back and answer one other question first. You asked about the frequency response of the pickup. It is good down to about five cps, and it is also fairly good up to 2,000 cps. It drops off in frequency response, but it is still quite useable. Actually, there was no significant vibration above 500 cps.

G. W. Dorr, Eng. & Res. Corp.: Did you have any trouble with the tape tangling?

McIntosh: No. This recorder went through extensive environmental tests before we ever put it in an airplane. It passed specification 41065B61 for vibration, which apparently was sufficiently severe so that the equipment gave us no trouble at all during the laboratory tests. We have no evidence that it did during gunfire, either.

Martin Drlik, Jack & Heintz: You mentioned a large amplitude of the voltage-regulator assembly within the inverter control box. Could you tell me how you measure amplitude of that particular component?

Kennard: We used a very small Gulton accelerometer to measure vibration inside the inverter control box.

Drlik: Was the accelerometer attached to the regulator?

Kennard: Yes.

Drlik: Can you state the maximum number of g's obtained?

Kennard: No, I cannot right now. I believe it was less than one g.

Drlik: At low frequency?

Kennard: Yes.

McIntosh: The motions, I mentioned, were of large amplitude and low frequency. The amplitude was 0.9 in. and the frequency around five to ten cycles.

Drlik: I understand that at 300 cps, the amplitude was about 0.004 in. That makes the acceleration approximately 18 g's. Did the acceleration ever exceed that value?

McIntosh: The acceleration that you mentioned is the maximum value we got from our magnetic tape. It may have been transient; we cannot tell for sure from our analysis.

Drlik: Did the inverter continue to function throughout the test? Was there any failure at all that you encountered?

McIntosh: There were some inverter failures, but I am not sure how many can be attributed to vibration, directly.

V. R. Boulton, Aerojet: At what point in the instrumentation system was the velocity signal integrated in order to get displacement?

McIntosh: We actually didn't perform electrical integration. We merely computed amplitudes from velocity and frequency.

Boulton: Did you record velocity directly?

McIntosh: Yes.

A. F. Dickerson, WADC: Is there any limitation imposed upon amplitude at the higher frequencies?

McIntosh: There is no limitation on voltage amplitude input. The recorder remains similar throughout the entire frequency range.

Dickerson: I mean as a result of using frequency modulation as a medium.

McIntosh: Apparently, no. The recorder works as well at high frequency as it does at low.

Dickerson: This question is somewhat aside from shock and vibration; it ties up with the recorder system itself, or more closely with the modulation index capabilities of the FM system. For a high frequency modulation, with a relatively low frequency carrier, there is some limitation on the amplitude of the modulating signal that can be put on the carrier. Inasmuch as the modulating signal frequency is quite high with respect to the carrier frequency itself, I am wondering if there wasn't an amplitude limitation as a result of this effect.

McIntosh: Yes. The voltage was accurate to ±5 percent when a constant-value sine wave was recorded.

D. F. Eldridge, Boeing: How about frequency accuracy?

McIntosh: It is much better. The frequency accuracy is probably as good as 1/10 of 1%, at 20 cps.

Eldridge: Do you use the same type transport mechanism in the airplane as in your analysis, or a different one?

McIntosh: We used the same type transport mechanism.

Dickerson: What were the net over-all frequency and amplitude response characteristics of the recording system, inasmuch as it involved FM?

McIntosh: The frequency response is practically flat, from 3 to 1600 cps. It has some tendency to drop off at 1600 cps. However, we don't use it beyond 2000 cps. You can put one volt into the system, and it will play it back faithfully at five cycles per second, or you can do it at 1600 cycles per second, and it does it just as well.

Dickerson: Is this voltage level well above what you encountered from the accelerometer?

McIntosh: Yes: The output of the accelercmeter is in milliwatts.

* * *

DESIGN OF CLOCK-TYPE MECHANISMS TO MEET EXTREME SHOCK AND VIBRATION REQUIREMENTS

J. W. Talcott, NOL

The effect of high-velocity water-entry shock, and transportation vibration on a clock-type mechanism and the practical solutions which eliminated failures, are discussed.

INTRODUCTION

This is a history of the testing that contributed to the development of a special-purpose clock with particular uses in naval ordnance. Known as Clock Delay Mk 17, the device provides delayed arming in a 2000-lb aircraft-laid mine. While the "case history" of tests on this piece of equipment is necessarily specific, there are a number of principles involved that are applicable to any design project.

The development of the clock began with a group of models, Clock Delay XD-1A, which for production purposes was modified from an existing mechanism. When the clocks as modified were put through a series of shock tests, a number of serious defects were revealed. This led to recommendations which were carried forward in a series of seven groups of working models. When the testing procedure was repeated and expanded, a satisfactory design was found for the clock.

TEST PROCEDURES

The considerations involved in the development procedure led to the establishment of some major parameters for the problem. These are illustrated in a series of five figures, with Figure 1 showing the size of the clock, and Figure 2, the

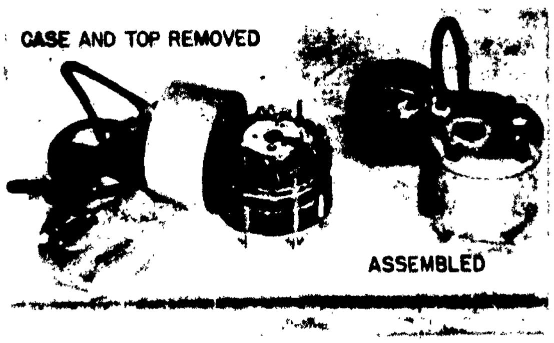

Figure 1 - Clock Delay Mk 17 Mod 0 for one-hour delayed arming of naval mines

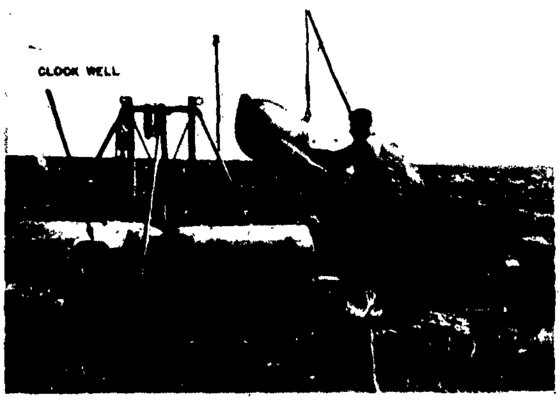

Figure 2 - Naval mine (showing clock well)

Figure 3 - Naval mine, showing damage resulting from water entry

size of the mine. The clock-well can be seen on the side of the mine case toward the nose. In Figure 3, the magnitude of the forces involved may be noted by observing the damage at the tail. The shock pattern as defined by an air gun shock test is shown in Figure 4. During these tests, which were conducted at -65°F and at room temperature, the peak acceleration of the impact phase was in the order of 5,000 to 8,000 g's. The definition of the shock pattern for the continuous two-phase shock by the air gun, is as follows:

1. The first (impact) phase consists of a velocity change of 18-20 ft/sec within 0.2 to 0.4 ms.

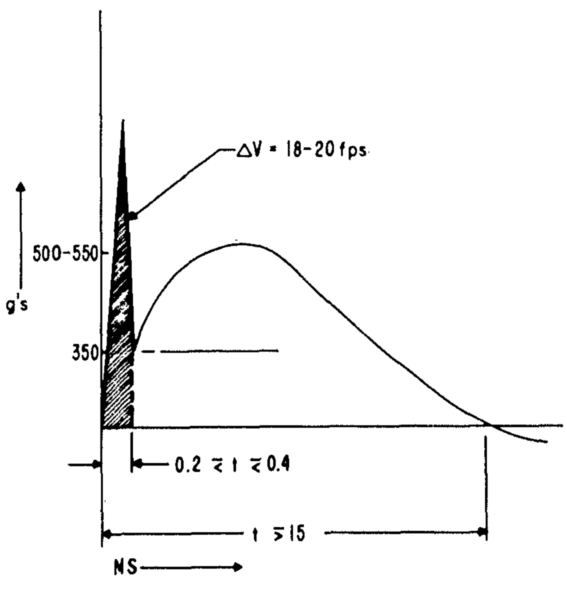

Figure 4 - Two-phase air gun shock test

2. During the second (drag) phase, the device is accelerated for a minimum of 15 ms in such manner that during this time the minimum average acceleration is 350 g's. The peak acceleration is between 500 and 550 g's.

A testing procedure of this type is often used for devices designed for use in mines (e.g., the Mk 39) to be dropped from 30,000 ft without a parachute. This drop results in a water entry velocity of 750-800 ft/sec. One characteristic of the test stems from the fact that it may be applied in two mutually perpendicular directions, one of which is the direction of most likely initial water entry.

TABLE 1
Vibration Recorded on Mk 39 Mine in Transit on 2-1/2-Ton Cargo Truck

Record	Direction: Fore and Aft	Crosswise	Vertical
30 mph on Macadam Road			
Av. Acceleration, g's	0.25	0.37	No Record
Max. Acceleration	0.37	0.50	
Min. Acceleration	0.16	0.21	
Av. Frequency, c/m	1,060	1,050	
Max. Frequency	1,360	1,550	
Min. Frequency	910	870	
20 mph on good Asphalt Road			
Av. Acceleration, g's	0.21	0.28	0.18
Max. Acceleration	0.27	0.50	0.24
Min. Acceleration	0.16	0.19	0.10
Av. Frequency, c/m	1,325	1,075	1,470
Max. Frequency	2,140	1,260	2,500
Min. Frequency	905	890	1,000

Table 1 is a sample of the vibration data recorded in the tests made on the Mk 39 mine during transportation by truck. Data are given for the fore-and-aft, crosswise, and vertical directions on two types of road.

A typical vibration table upon which these mechanisms were given the simulated vibration test is shown in Figure 5. The mechanism was rigidly fastened to the table of the vibrator in mounts simulating service mounts. To obtain the most significant directions of the vibration relative to the mechanism, the device was mounted several times in different orientations. Tests were made at -65° F, +160° F, and at room temperature in accordance with the following schedule, based on single amplitude (zero to peak) and frequency:

Amplitude (In.)	Frequency (cpm)
0.030 (± .002)	700-1500
0.020 (± .002)	1600-2000
0.013 (± .001)	2100-2500
0.009 (± .001)	2600-3000

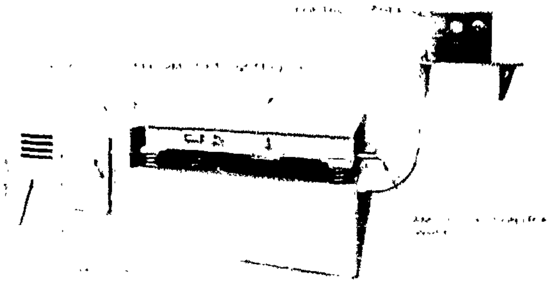

Figure 5 - NOL vibration testing equipment, Type 1B

The frequency was changed in increments of 100 cpm with the duration of each frequency step one hour when the mechanism was mounted in only one orientation. When it was mounted in more than one, the hour was divided among the several orientations. The test resulted in a rigid-body peak acceleration as great as 2.3 g's. It took 24 hours to make the test.

CORRELATING TEST RESULTS TO DESIGN

A study and analysis of the tests began with the observable failures and went on to point out remedies. The major failures by types were:

- Bent mounting studs
- Bent timing staffs
- Fractured internal spacing studs
- Cracked plastic caps.

In the case of the mounting studs, due to interchangeability it was not feasible to increase the diameter and so strengthen the stud. However, the material could be changed with the result that the mass of the clock would be reduced. Consequently, the studs were changed from brass with a yield point of approximately 25,000 to an aluminum alloy with a yield point of approximately 65,000. At the same time, all possible internal brass parts were changed to aluminum, and lightening holes were added (Figure 6). The weight reduction was about one-third in the clock as a whole. In this change, the plastic case top was replaced by an aluminum top.

Figure 7 shows how a spacer was substituted for the nut between the top plate and the contact plate. As a result, the threaded section of the stud was lengthened 3-1/2 times and gained that much in which to stretch and absorb energy.

Attempting to correct for the bent timing staffs, it was reasoned that some equivalent static force might be determined which could be used in cal-

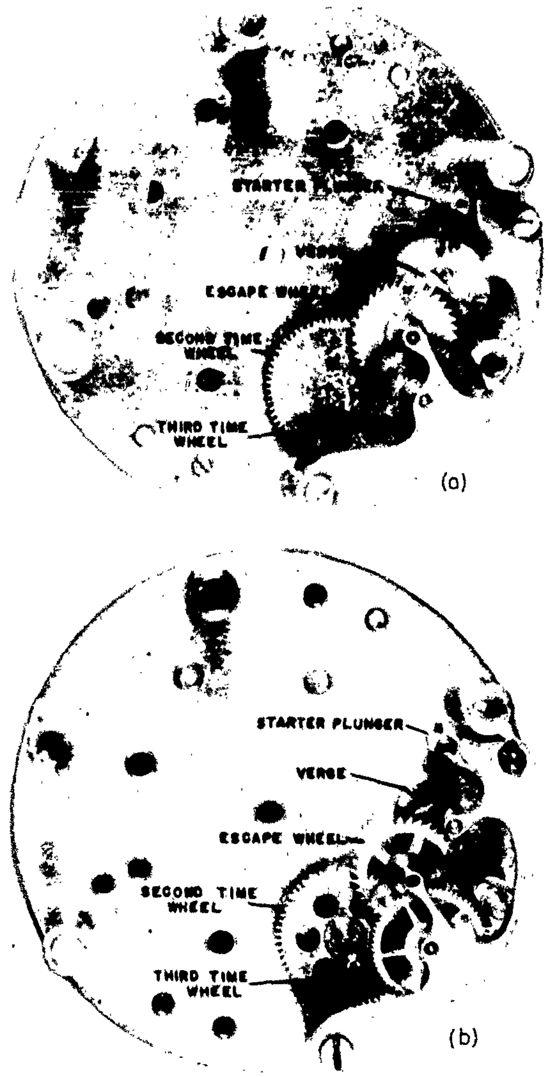

Figure 6 - Modification of original clock
(a) Clock Delay XD - 1A
(b) Clock Delay Mk 17 Mod 0

culating the stress in these shafts. Starting with either $F = (W/g)a$ or $K.E. = 1/2\,(W/g)\,V^2$, it may be shown that the force felt by the staffs is $F = kW$, in which k is a variable constant. A value of 2000 for k was found to give satisfactory stresses, probably in the region of one-half the yield point. In essence, this is equivalent to saying that the force felt by the mechanism, due to the two-phase shock, is that which would be produced by an acceleration of 2000 g's.

The required changes in the design of the clock and of material for construction of the vital components as determined by the testing procedure

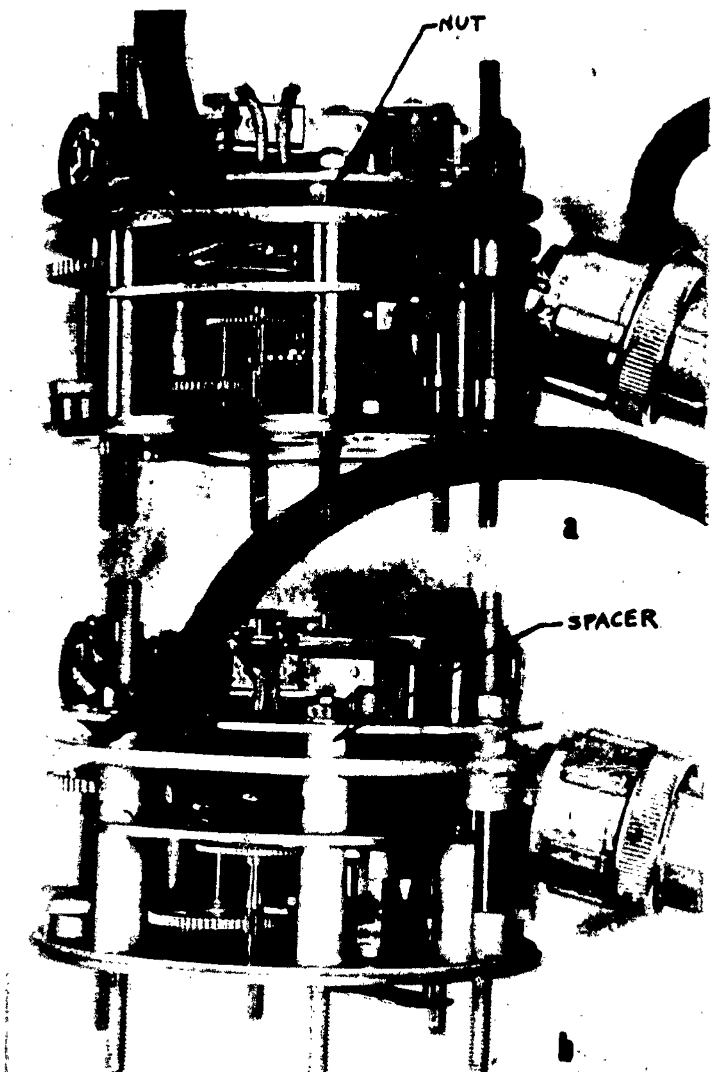

Figure 7 - Modification of original clock
(a) Clock Delay XD - 1A
(b) Clock Delay Mk 17 Mod 0

previously discussed, led to the development of the device known as Clock Delay Mk 17. When this clock was subjected to the same shock and vibration tests as were given the experimental models, there were no damage effects of a serious nature. The proper tightening torque on all screws was found to be mandatory. Switch chatter had been eliminated. Although it did not cause trouble in this design, the brinelling effect of staff pivots vibrating against the sides of the pivot holes has caused trouble in similar mechanisms. The original cardboard box packaging for the clock was not satisfactory. There was excessive chafing of the clock cable against the cardboard. Packaging of the clock and associated cable in a sealed can has proven satisfactory.

Two major points of a slightly different nature arose during the evaluation of this clock. The first point is the necessity for the design engineer to completely subordinate his engineering pride during the development phase. Many engineers are very delighted when one of the first engineering models of their design actually works. They apparently forget that it is just as important that the model should also work after shock and vibration evaluation. The second and most important point is the need for shock and vibration

evaluation during the development phase. The Naval Ordnance Laboratory is very fortunate in having the Janus two-phase air gun and the evaluation engineers who guide the testing. The effects on development models of the simulated shock and of the actual aircraft drops were practically identical. During the development phase of this project, there were seven groups of models, with progressive improvements incorporated as a result of shock and vibration evaluation of each group, before a satisfactory design was obtained. It is hoped that the impression is not conveyed, due to the theoretical work in previous paragraphs, that the design of this clock was accomplished entirely on paper. The fact that we can now feel a little more confident in designing new mechanisms for shock and vibration is due solely to the evaluation work during the development phase.

In conclusion, the author wishes to acknowledge the contributions of S. J. Black, co-designer on this project, and the splendid work and cooperation of the personnel of the Technical Evaluation Department of NOL.

DISCUSSION

J. Steinman, Hughes Aircraft: Why was the clock made of brass in the first place?

Talcott: Most of our clocks are made of brass; it is an old clock custom. We purchase our clocks; the Gun Factory does not make them. These clocks are obtained from private industry, and are not manufactured for the type of service required in this application of the delayed arming of aircraft-laid mines. As I said before, we started out with a group of engineering models. These models were made by modifying as slightly as possible the clocks which we had. When these instruments are to be mounted in mines which use parachutes to reduce the water-entry velocity, they are usually made of brass and are satisfactory. Material, other than brass, which will give a reduction in weight may be used. However, weight is not too much of an item with us, especially the weight of such a small component as a clock. I might say that the clocks which we are developing are almost entirely of the high strength 75 ST aluminum.

K. G. F. Moeller, USNEES: Referring to the last picture you showed, is it correct that maximum deceleration is in the plane of the screen and that the maximum force is perpendicular to the shaft?

Talcott: That is correct.

Moeller: When the force is applied perpendicularly to the shaft, the bending is greater.

Talcott: It is possible that the clock may land in a better orientation; but it can land in this orientation, which is the basis on which the design was done. That is, the forces are perpendicular to the axis of the shaft.

Dr. H. T. Wensel, Chairman: Is the device mounted in the orientation required to give forces normal to the longitudinal axes of the shafts?

Talcott: Mounted that way in the mine? No. This is not a parachute-controlled mine. Ballistic properties are poor. The mine is liable to land in quite a few different orientations.

Moeller: What type of bearing do you use in the clock for the shaft?

Talcott: These bearings are simply the pivot on the end of the staff riding in a hole in the aluminum. This is only a one-hour clock. It is not one of our more complicated longer-time clocks.

F. C. Smith, NBS: Do you have any design data to indicate the magnitude of the acceleration when the mine hits the water?

Talcott: Yes, we do. I have often thought that our original design models acted as accelerometers for the evaluation of the forces in the mine resulting from the water-entry shock. The data was most evident, to one of experience. The parts that yielded, the amount they yielded and the actions of the components were such as to give a fair value for k in the expression F = kW.

Smith: Did you find any correlation between the failures you had in the field and those which you were able to produce in the laboratory?

Talcott: The data were very similar. The shaft yield at the midpoint, that is, the shaft eccentricity, was 0.0002 in. after simulated shock test, as compared to 0.0002 to 0.0003 in. after airdrop. The cracking of the plastic caps was identical.

The fracturing of the studs was identical. The amount of bending in the mounting studs was identical. I think it was an excellent vindication of the use of the two-phase air gun.

Dr. W. J. Sette, DTMB: You said you risked a few calculations using the constant acceleration method. What value of g did you use?

Talcott: We started off with the value, 1000 g's. As I mentioned, the peak acceleration is between 4000 and 7000 g's. It would be very difficult to design for the peak, so we started with 1000. This showed improvement, but some bending of the shafts was observed. No bending can be tolerated; if the clock shaft turns eccentrically, the mine may become a "dud." The bending was observed in one of the groups of seven models previously mentioned. Then we went to 1500. Finally, in order not to have any bending that could be detected with a toolmakers' microscope, a value of 2000 g's was chosen. This value, which makes F = 2000 W, gave stresses between 10 and 90 percent of the yield point for the components.

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THE DEVELOPMENT AND USE OF A STATISTICAL ACCELEROMETER

P. R. Weaver, NBS, and R. N. Evenson, BuAer

A reliable and compact instrument for recording the occurrence of maneuvering loads on aircraft is described. Sensing elements record the number of times predetermined accelerations occur. Methods of using the data in the preparation of design specifications and in the prediction of the life of aircraft are discussed.

INTRODUCTION

The development and tests of a statistical accelerometer for the Bureau of Aeronautics, Department of the Navy, are described. The characteristics of this instrument make it suitable for counting the occurrence of unidirectional acceleration maxima of chosen magnitude relative to the steady flight acceleration of 1g.

For many conditions of flight, and particularly for maneuvering loads, it is well established that the vertical acceleration at the center of gravity is proportional to the stresses in the primary structure of the wing. Where this is true, measurement of the number of times different levels of acceleration are attained at the center of gravity gives valuable data on the number of cycles of different levels of stress in the wing.

In pursuing the problem of determining the service life of aircraft structures as related to fatigue, it becomes desirable to know first the so called S-N curve for the airplane and second, the history of loadings experienced by the aircraft under typical service conditions. The need for a history of actual service loading has resulted in the development of the statistical counting accelerometer. Certain basic requirements had to be met in the design of the counting accelerometer to make it acceptable for field service use. These are:

(1) Small volume and light weight,
(2) direct recording and reading,
(3) low maintenance,
(4) high reliability,
(5) ability to operate from the existing aircraft electrical system (in this case 28 v, dc).
(6) Satisfactory response - must faithfully sense and record all maneuvering loads experienced by an airplane.

DESCRIPTION OF INSTRUMENT

The accelerometer, Figure 1, is composed of four measuring and recording units. This number of units may be varied so as to count acceleration maxima at any desired number of acceleration levels. Each channel consists of an acceleration-sensitive switch as shown in Figure 2, and its related resistor, diode, and electromagnetic counter, as shown in the circuit diagram, Figure 3.

The acceleration-sensitive switches are so connected in the circuit that they prevent the counters from recounting until the acceleration has returned to a value less than that of the first sensing element. This is achieved as follows: when the reference acceleration switch closes to make the first counter respond, it also applies to the remaining counters, through 510-ohm resistors, a "holding" voltage. This "holding"

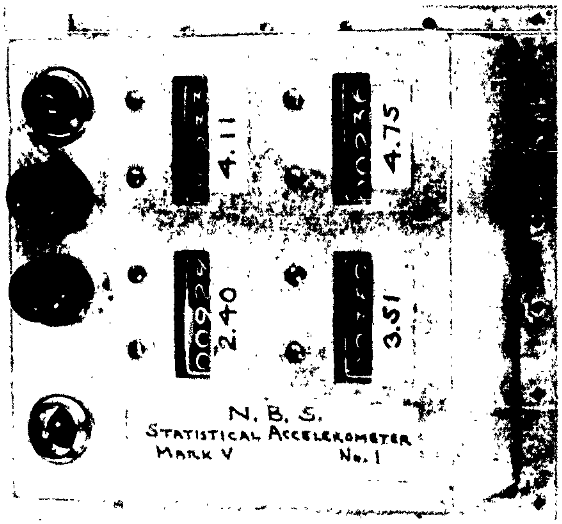

Figure 1 - NBS Statistical Accelerometer Mk V

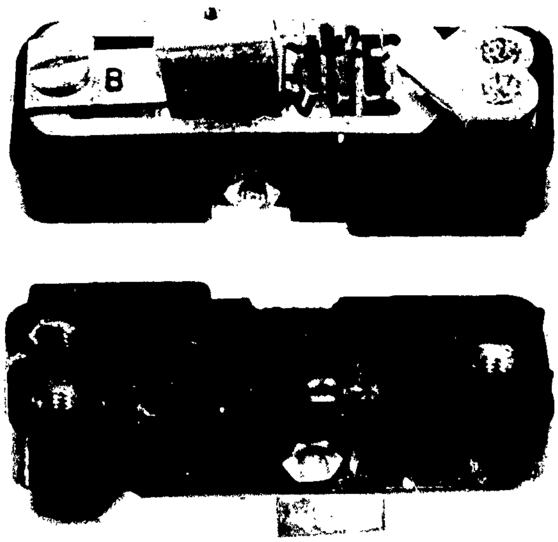

Figure 2 - Acceleration-sensitive switch

voltage is insufficient to actuate the remaining counters by itself, but is sufficient to hold them in a cocked position once the appropriate

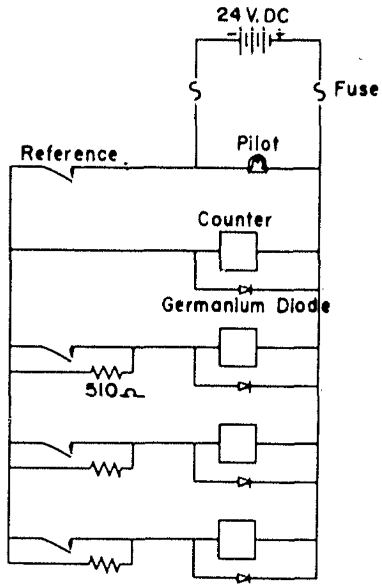

Figure 3 - Schematic diagram of NBS Statistical Accelerometer Mk V

acceleration switch has closed to make them count. Only after the "holding" voltage is removed and the counter is released from a cocked position, can the counter be indexed another unit. Figure 4 shows a typical acceleration-time curve

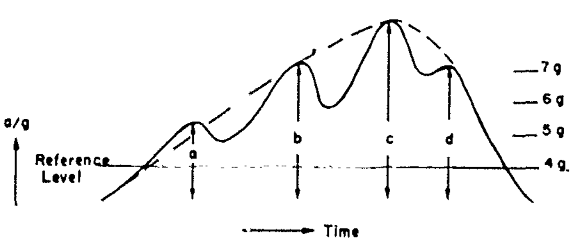

Figure 4 - Typical acceleration-time curve of a maneuver

for a maneuver. Without the reference level switch the individual peaks a, b, c, and d, could be of sufficient value to cause the instrument to record each peak when the only acceleration of importance, load-wise, would be the maximum value indicated by the dotted line. The snap action of the acceleration switches causes them to release at acceleration levels about 25 percent below those required to make them close. The "holding" voltage from the reference acceleration switch will therefore not be removed until the acceleration level has dropped to about 25 percent less than that required to close the reference switch.

DETERMINATION OF NATURAL FREQUENCY

The natural frequency was determined by mounting a switch on a Model 6 Calidyne electrodynamic shaker and sweeping the frequency band with a constant oscillator output. This was repeated for three nominally identical switches and four masses whose weights differed less than 25 percent. The results of these tests are given in Table 1. It can be seen that all the natural frequencies lie between 40 and 55 cps. The vibration consisted primarily of the mass-spring system vibrating as a cantilever beam.

TABLE 1
Undamped Natural Frequency
of Unimax Acceleration-Sensitive Switches

Sample No.	Natural Frequency (cps)			
	With Mass a	With Mass b	With Mass c	With Mass d
1	--	45	--	--
3	--	--	--	55
5	42	--	40	45

DYNAMIC AND STATIC CALIBRATION

The spring blade of the switch, B, Figure 2, has a slot in which the mass is fastened. By moving the location of the mass along this slot, the effective spring length is varied. This allows simple adjustment of the g-level setting of an element.

Each switch sensing element was adjusted and statically calibrated on a horizontal centrifuge. Then, the element was mounted on an electrodynamic shaker and its response to sinusoidal acceleration determined. The accelerations required to close the switch were determined from measurements of the output of a barium titanate accelerometer mounted on the shaker. A cathode-ray oscilloscope in conjunction with an oscillator was used to indicate closure of the switch.

The results of the tests are given in Table 2 and Figure 5.

To determine what effect a counter would have on the response, another switch was mounted on a double cantilever beam. In these tests the beam was deflected just enough so that when released it caused the switch to index the counter one unit. The acceleration experienced by the switch was then computed from the measured frequency and deflection of the beam. The results of these tests are given in Table 3 and Figure 6.

TABLE 2
Static and Dynamic Calibration of Switch

Frequency (cps)	a/g	
	Static	Dynamic
0	4.8	
6.58		5.2
14.0		4.9
16		5.1
20		5.1
22		5.1
24		5.2
26		6.6
28		6.6
30		6.6
35		4.7
40		2.7
45		.9
50		1.4
60		5.6
70		9.4
80		10.7

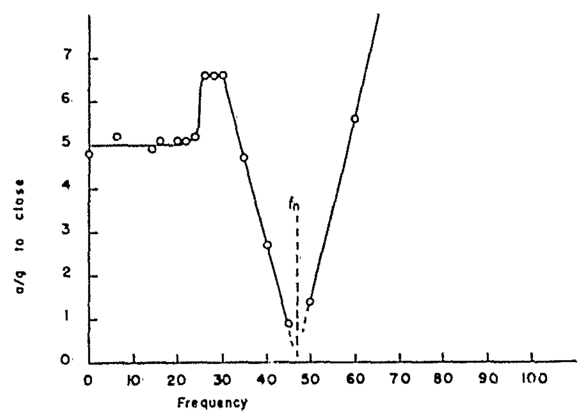

Figure 5 – Frequency response of switch

FLIGHT TESTS

Installation of instruments for tests on an F6F airplane was as follows:

TABLE 3
Static and Dynamic Calibration
of Switch with Counter

Frequency (cps)	a/g		a/g Dynamic / a/g Static
	Static	Dynamic	
0	5.6		1.00
3.33		5.5	.98
4.29		5.4	.96
5.68		5.5	.98
8.00		6.7	1.20
11.1		8.9	1.60
17.4		12.6	2.20

Note: Above 20 cps, the counter will not index when switch is subjected to approximately 15 g's.

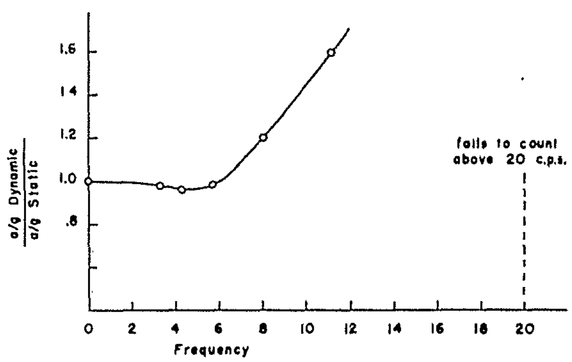

Figure 6 - Frequency response of switch with counter

The statistical accelerometer was mounted approximately four feet aft of the center of gravity of the airplane. A maximum-indicating accelerometer and a V-G recorder were mounted at approximately the center of gravity and their readings were compared with the results obtained from the statistical accelerometer. The aircraft power supply was used to operate the statistical accelerometer.

Before each flight, readings of the statistical accelerometer were taken and a new slide was inserted in the V-G recorder. The pilot noted the maximum acceleration recorded by the maximum-indicating accelerometer and reset the instrument after each maneuver. After each flight the V-G slide was removed and processed and the readings of the statistical accelerometer were recorded.

Table 4 gives the results of several flights. For these flights, the statistical accelerometer was turned on after take-off and off before landing; thus, only accelerations experienced during flight were recorded.

TABLE 4
Maneuvering Flight Data

Flight	a/g	Statistical Accelerometer				VG Recorder n	Max. Ind. Accelerometer n
		Readings		N*	n'*		
		Before	After				
1	2.40	455	463	8	2	2	2
	3.51	17[illegible]	184	6	2	2	2
	4.11	[illegible]	98	4	2	2	2
	4.75	178	180	2	2	2	2
2	2.40	463	472	9	3	2	2
	3.51	184	190	6	2	2	2
	4.11	98	102	4	2	2	2
	4.75	180	182	2	2	2	2
3	2.40	472	519	47	40	-	-
	3.51	190	197	7	2	-	-
	4.11	102	107	5	2	-	-
	4.75	182	185	3	3	-	-

*N is the total number of times the a/g level is equaled or exceeded, and n' is the number of times the a/g level is equaled or exceeded and the next higher a/g level is not.

Table 5 gives the results of a flight in which the statistical accelerometer was turned on before take-off and kept on while the pilot made four landings. In this type of test the accelerometer was subjected to moderate frequency, low acceleration excitation. As is seen from Table 5, no false counts were obtained. Since the accelerometer was found to be insensitive to this excitation, it was left turned on during take-off and landing for subsequent flights.

TABLE 5
Landing Data

a/g	Statistical Accelerometer				VG Recorder n	Max. Ind. Accelerometer n
	Readings		N	n		
	Before	After				
2.75	78	78	0	0	0	0
4.02	21	21	0	0	0	0
5.10	25	25	0	0	0	0
6.16	115	115	0	0	0	0

To determine the effects of extended flight service on the statistical accelerometer, it was left installed while the airplane completed about fifty hours of operational training flights. Table 6 gives the resulting data. The instrument was then recalibrated on a horizontal centrifuge. Table 7 gives the results of this test, showing the change in calibration. Figure 7 shows the frequency of occurrence of various acceleration maxima recorded during the fifty hours of flying.

TABLE 6
Operational Training Flight Data (Statistical Accelerometer)

Date	2.40 g's			3.51 g's			4.11 g's			4.75 g's		
	Reading	N	n	Reading	N	n	Reading	N	n	Reading	N	n
	519			197			107			185		
12-18	584	65	53	209	12	6	113	6	5	186	1	1
12-19	650	66	47	228	19	4	128	15	13	188	2	2
12-20	690	40	31	237	9	9	128	0	0	188	0	0
12-22	692	2	2	237	0	0	128	0	0	188	0	0
12-27	695	3	3	237	0	0	128	0	0	188	0	0
12-30	695	0	0	237	0	0	128	0	0	188	0	0
1-5	724	29	14	252	15	13	130	2	2	188	0	0
1-6	728	4	3	253	1	1	130	0	0	188	0	0
Total for 50 hours		209	153		56	33		23	20		3	3

TABLE 7
Calibration of Accelerometer After Fifty Hours of Flight

Switch	a/g Setting		a/g
	Original	Final	
1	2.40	2.50	+0.10
2	3.51	3.56	+0.05
3	4.11	4.14	+0.03
4	4.75	4.89	+0.14

DISCUSSION

The results obtained on the switch and counter unit, as shown in Figure 6, indicate that the response deviates sharply from a straight line at approximately 6 cps. This is a desirable feature since high frequency loadings, which in general contribute little to fatigue damage, will not be recorded.

The statistical accelerometer is capable of furnishing data useful in the prediction of the fatigue damage due to repeated maneuvering or gust loads on a fighter airplane. Accelerations due to engine vibration, gun firing, buffeting, landing and take-off are all beyond the frequency range or below the acceleration range of the instrument. Their contribution to the fatigue of the primary wing structure of a fighter airplane is considered to be relatively small.

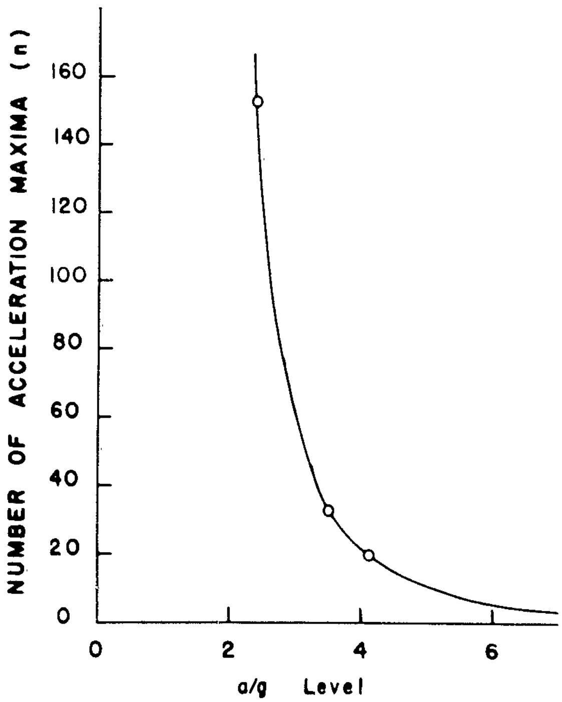

Figure 7 - Occurrence of acceleration maxima experienced by an F6F during 50 hr of operation training flights

Additional development of statistical accelerometers is needed to separate these acceleration counts into groups according to the airspeed and altitude at which they occur. This information would supply a basis for recommendations on operating speeds and altitudes.

APPLICATION

It has been proposed to apply Miner's theory of cumulative damage to the over-all structure of an airplane in order to predict the percent of fatigue life remaining in the airplane. To apply this theory, two sets of information must be known: first, the S-N curve for the airplane structure, and second, a detailed record showing the history of loadings that a particular airplane has experienced. The S-N curve can be obtained by performing repeated load tests on several structural specimens at different load levels. Figure 8 gives the S-N curve resulting from such tests on a series of AD airplane wing panels. This curve is based upon four repeated load tests at different load levels and one static test which has its point at a single application of load. The history of loadings is obtained by installing a counting accelerometer in an airplane in actual service use.

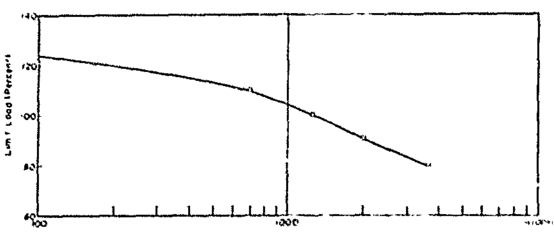

Figure 8 – S/N curve for a series of AD airplane wing panels

Miner's theory is based on the hypothesis that each individual loading on a specimen contributes to the eventual failure of that specimen. Although Miner's original work was performed primarily on simple specimens, it is possible that the theory will apply to complex structures such as airplane wings. This theory can be expressed as the equation

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \ldots = 1.$$

The denominators represent the total number of loading cycles which at a particular load level, cause failure of the specimen. These values are therefore points on the S-N curve of the specimen. The numerators represent the number of cycles experienced by the airplane structure at that particular load level. If the specimen has failed under loadings represented by all the numerators, the sum of these ratios will be unity. If the specimen has not failed under these loadings, the sum will be less than unity and will represent the proportion of total life which has been consumed. Figure 8 and Tables 8 and 9 show how this theory might be

TABLE 8
Application of Miner's Theory to AD Airplane

Percent Limit Load	Accelerometer Settings a/g	Total Number N	Number of Times Acceleration Occurred for h hours Flying Time $n_{\%LL}$
80	5.6	$60 = (N_{80})$	$10 = (N_{80}-N_{90}) = n_{80}$
90	6.3	$50 = (N_{90})$	$36 = (N_{90}-N_{100}) = n_{90}$
100	7.0	$14 = (N_{100})$	$12 = (N_{100}-N_{110}) = n_{100}$
110	7.7	$2 = (N_{110})$	$2 = (N_{110}) = n_{110}$

TABLE 9
Application of Miner's Theory to AD Airplane

Percent Limit Load	a/g Corresponding to % Limit Load	Percent Damage per Cycle from S-N Curve $D_{\%LL}$	Average Damage for Range $d_{\%LL}$
80	5.6	$0.028 = (D_{80})$	$0.038 = \left(\frac{D_{80}+D_{90}}{2}\right) = d_{80}$
90	6.3	$0.049 = (D_{90})$	$0.064 = \left(\frac{D_{90}+D_{100}}{2}\right) = d_{90}$
100	7.0	$0.080 = (D_{100})$	$0.111 = \left(\frac{D_{100}+D_{110}}{2}\right) = d_{100}$
110	7.7	$0.149 = (D_{110})$	$0.4 = d_{110}$*

$$\text{Percent Life Used} = K\left[n_{80}d_{80} + n_{90}d_{90} + n_{100}d_{100} + n_{110}d_{110}\right]$$

$$= 2.0\left[(10)(0.038) + (36)(0.064) + (12)(0.111) + (2)(0.4)\right]$$

Percent Life Used = 9.7

* Based on estimated frequency of occurrence of loads above 110% limit load.

applied to the AD airplane. The values of percent damage per cycle, $D_{\%LL}$ (in Column 3, Table 9), were obtained by picking off values of N from the S-N curve, Figure 8, corresponding to the load levels shown in Table 8 and then taking their reciprocals. This is in accord with the contention of Miner's equation.

Since the data obtained from the counting accelerometers represent the number of occurrences in the ranges from one load level to the next, the average percent damage in each range, $d_{\%LL}$ (in Column 4, Table 9), must be used. For the percent damage above the highest load level, a higher level must be estimated beyond which there is no likelihood that loadings will occur. The factor K shown in the equation given in Table 9 accounts for the percent damage contributed by loadings below the level recorded by the counting accelerometer. The summation of the products of percent damage per cycle and the actual number of loadings experienced as given by Table 8, gives the percent of life expended by that particular airplane. The proof

that Miner's theory can be applied in this manner to structures as complex as an airplane wing has yet to be established. The Bureau of Aeronautics is currently performing repeated load tests on wing panels from an F9F airplane. These tests should provide some indication of the validity of applying this theory to complex structures. In a recent test of one wing specimen under repeated loads at various levels, the application of Miner's equation to the loading history to failure and the basic S-N curve of the wings, resulted in a value of approximately 1.15, which shows some tendency towards correlation.

This illustration is only one application for which the counting accelerometer might be used. Another application is the gathering of a detailed loading history from a particular type of airplane. This can be accomplished by installing counting accelerometers in all the airplanes of a given type in one or more squadrons. The data obtained would then be representative of typical squadron operation for the airplane. Such data can then be used for consideration of design requirements and specifications for new airplanes that have a similar operational mission.

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DISCUSSION

Dr. M. G. Scherberg, WADC: You stated that the use of Miner's theory still has to be established for complex structures of the airplane wing or other structures on the airplane. It is my impression that the theory still has to be established even for simple structures.

Evensen: That is probably true if based on Miner's paper. He certainly did not show that $n_1/N_1 + n_2/N_2 + \ldots$ always equaled one. But, as I said, we did get one test which checked fairly close, even closer than some of Miner's. Of course it is just one test; we hope that additional tests at different load levels and with different histories will bear it out.

Scherberg: It is a nice theory.

Evensen: Very nice, very convenient, very simple.

R. N. Janeway, Chrysler Corp.: We have had a lot of experience in the use of this type of accelerometer in railroad testing over many years, and I think it is an ideal instrument for that type of field testing. We have always found, however, that it was necessary to damp the elements, using viscous damping, because otherwise we tend to get scatter. We use a combination of viscous and Coulomb damping with a timing delay in the circuit. I wondered what expedients you have had to resort to.

Evensen: Mr. Weaver, who is the developer of the sensing element, provided damping in the sensing element by putting two metal plates on each side of the cubical mass. By dropping viscous fluid in between the plates and the mass, very favorable damping characteristics were obtained. However, we found that the natural frequency was up around 40 cycles for the sensing element itself. Since we are limited by the counter, Mr. Weaver found that we could get by without using damping for our particular application. But he has made provision for it for such purposes where it is found necessary.

Weaver: I was wondering if you are familiar with the work of the Royal Aircraft Establishment in England of checking Miner's ratio for the total structure itself. The work done at RAE has shown that Miner's ratio seems to hold very well for wing structures, the reason being that you have a stress concentration factor of four, whereas in a specimen itself you don't know what you have. Unless the edge conditions in the specimens that you are checking in fatigue are all identical, you are going to get appreciable scatter. Likewise you have certain concentrations in the material itself that will contribute to scatter. I think you will find much better agreement in structures than you will in any type of specimen.

* * *

CONCEPTS OF SPECIFICATIONS IN MOBILE MILITARY GEAR

V. V. Gunsolley, BuAer

A cursory survey of vibration tests on airborne equipment, as currently specified by various agencies, reveals widely differing requirements for essentially similar applications. An attempt is made to present a rational approach to vibration test specifications and to apply the power method of expressing vibration test requirements.

Vibration tests as currently specified in both general and particular specifications, indicate a somewhat indefinite understanding of vibration, and often a rather inconsistent approach to a determination of the parameters the vibration tests should have. The customary procedure is to specify either the acceleration or the excursion (double-peak amplitude) of the vibration over a desired frequency range that either experience, or just plain guesswork dictates. Not infrequently the specification varies the excursion and frequency range in somewhat arbitrary steps, thus breaking up the over-all frequency range into blocks and applying either a constant acceleration or a constant excursion, or both to each block.

Example of one of the more rational specifications:

Frequency Range (cps)	Requirement
10-28	0.25 in. excursion
28-150	10 g's acceleration
150-210	0.0088 in. excursion
210-2000	20 g's acceleration

This improvised type of specification may result from the intuition that since the acceleration increases to fantastically high values at high frequencies (with constant excursion) perhaps the excursion should be reduced on the higher blocks of frequency. Again by the same reasoning, if the number of g's is held constant over the frequency range, then as the frequency increases the excursion decreases to excessively small values, to satisfy the equation

$$G = 0.0514\ EF^2. \tag{1}$$

Where

G = acceleration (g's)

F = frequency of the vibration (cps)

E = excursion of the vibration (in.),

and from which

$$F = 4.41\ G^{\frac{1}{2}}\ E^{-\frac{1}{2}} \tag{1a}$$

and

$$E = 19.45\ GF^{-2}. \tag{1b}$$

This kind of specification results in marked discontinuities, as may readily be seen from a graphical plot of the resulting acceleration, excursion, and power.

From these equations only, any notion of what is taking place during vibration might still be somewhat vague. The specification writer, having no other means of specifying vibration tests, usually specifies both ways, as above. Vibration pickups can be installed at significant points of typical aircraft and the maximum excursion of the vibrations measured at the various frequencies encountered in actual flight. From these data, a composite aircraft curve of excursion versus frequency may be drawn (Figure 2) which will show the maximum excursion at each frequency, the equipment will ever be asked to withstand. This composite aircraft excursion spectrum may be multiplied by a safety factor and specified as a vibration test for any equipment that is to be installed in any of the aircraft analyzed. This is all very well but still does not give a readily discernible picture of the magnitude of vibration. Consequently, the designer of vibration test machinery may be left in somewhat of a quandary as to how to design a machine to meet general vibration test specifications, except in an entirely arbitrary manner. Once a machine is designed to meet an experimentally determined vibration test, it may become largely outmoded if, for any reason, the test needs to be greatly altered. This lack of a rational means of specification has in the past led to the appearance of a random variety of arbitrary designs of vibration test machines, few of which may meet the general requirements of vibration testing.

Continuing to think about the incongruities and marked discontinuities of such improvised vibration tests, it appears that the most likely element determining the effectiveness of a vibration test is the power of the vibration, rather than the acceleration, excursion, or frequency. This is analogous to the electric motor, where we could express the rating in terms of torque and speed, or in terms of volts and amperes and power factor rather than in terms of power. These factors would determine the power, but the power would need to be calculated before a reliable conception of what was going on in the circuit could be obtained. In vibration problems, acceleration, excursion, frequency, and even energy are powerless. Why not then express vibration in terms of power, independently of the powerless components?

With this objective let us consider Figure 1. A reference particle p moves in a circle at a constant angular velocity ω. The particle is projected to diameter E (equal in this case to the excursion) to the point M, at which is a 1-lb mass. The mass vibrates over E with a frequency of F cps, equal to the number of revolutions per

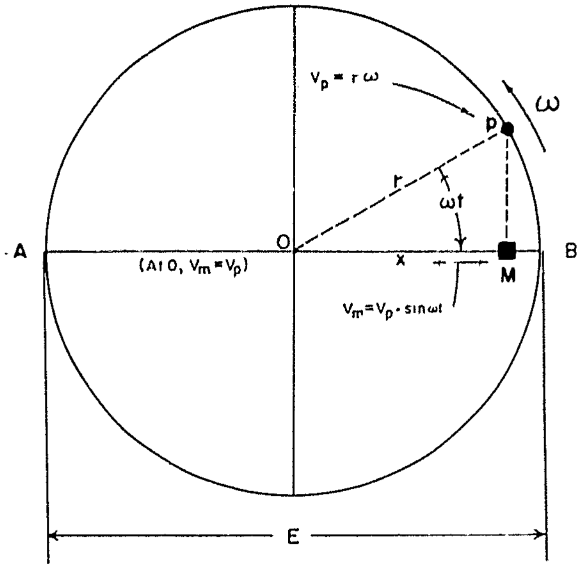

Figure 1 - System for production of simple harmonic motion

second of p. When both M and p are at B, no work is being done on M because it is not in motion. The acceleration a toward point A is a maximum here; hence $f = Ma$ is a maximum. As soon as M moves toward A an element of distance dx, an element of work f·dx is performed; since this is done in an element of time dt, an element of power is generated equal to f·dx/dt. When this instantaneous power is integrated over a quarter cycle, from B to 0, the average power is found to be

$$P = 5.78\ E^2\ F^3\ 10^{-3}\ \text{watts/lb.} \qquad (2)$$

This is the average power required during the first quarter cycle, to accelerate the mass M to the velocity ω_r which it has at point 0. If left to itself, the mass would continue to move in a straight line forever. However, it is retarded by the driving mechanism and the process reverses itself during the second quarter cycle. The stored energy in the mass is returned to the driving mechanism (flywheel or motor) while it is being slowed down to a stop at point A. Equation (2) gives the power associated with the energy that surges into and out of the mass M, at each stroke of the vibrator. If the system is ideal, the average power exerted on M in accelerating it from B to 0 is equal and opposite to the average power exerted by M on its driving machine when decelerated from 0 to point A. This power, though wattless, is, nevertheless, a measure of the vibration power to which M is subjected, and

which it must alternately store and deliver at each stroke of the vibration, without either breaking down or malfunctioning. If the mass absorbs power due to deformation or other cause, the power returned to the driver will not equal the power put in. The difference will not be wattless and will, in general, represent destructive and/or damping power. This will make no difference in the power of the vibration since the difference will be supplied by the driving motor and may be indicated by a wattmeter. The only power drawn from external sources is that taken by the losses in the driving machinery and in deformation of the test specimen. Thus, even if Equation (2) does not give all the power in the test specimen, it does give all the vibratory power therein. Neither the damping power, nor the destructive power, is peculiarly vibratory power, though modulated at double the vibration frequency. The vibratory power is modulated at four times the vibration frequency.

One current specification calls for a vibration test as follows: "The equipment shall operate satisfactorily when subjected to continuous vibration having a double amplitude of 0.01 in. at any frequency within the range of 5 to 500 cps and resulting in a maximum acceleration of 10 g's whichever is the limiting value." From Equation (2), the power is seen to vary as the cube of the frequency. Since the frequency variation is 100 to 1, it follows that the power ratio in this case would be a million to one if the acceleration were not limited to 10 g's. When acceleration is limited to 10 g's the excursion is reduced above 140 cps, but the power ratio below 10 g's is still 22,000 to one.

When we look at a composite aircraft vibration test curve on a log-log graph (Figure 2) with

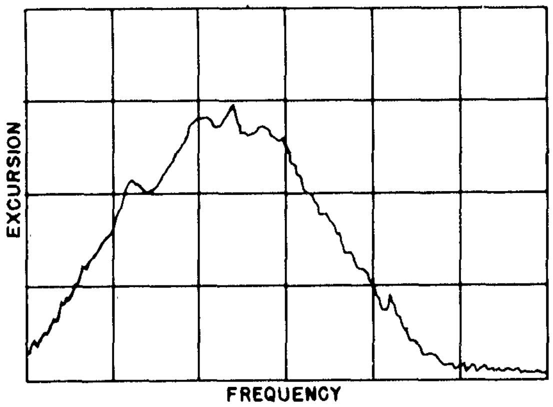

Figure 2 - Composite aircraft vibration test curve (log-log plot)

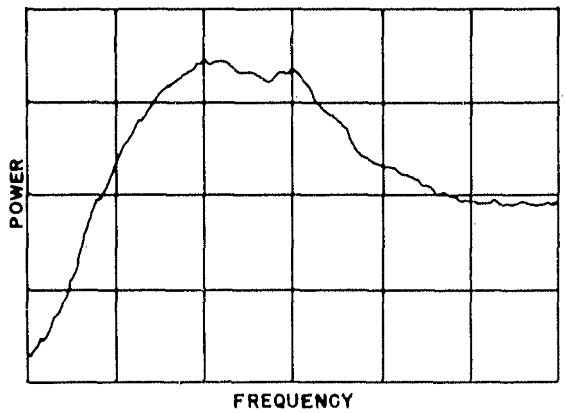

Figure 3 - Power spectrum obtained from Figure 2 by cubing the frequency and multiplying it by the square of the excursion

excursion E and frequency F as coordinates, we get no readily discernible conception of what is really taking place until we cube the frequency and multiply it by the square of the excursion, to get the power curve, Figure 3. This curve does not give power as a function of frequency, but the statistical power at any given frequency in the sense that no operations need be performed on the curve to get its meaning. Further, only one kind of test run over the frequency range is needed. It is no longer necessary to run first one test of constant G versus frequency and then another of constant amplitude versus frequency to be sure one has done the job thoroughly. If constant power is desired over the frequency range, it is necessary only to set in an automatic feed to the machine that will vary the excursion inversely as the 3/2 power of the frequency. This should be done by means of an interchangeable cam, since it may not be desirable always to test with constant power. For instance, if Figure 2 is converted to a power-versus-frequency curve, it may give a power spectrum something like Figure 3. This power spectrum is multiplied by the desired safety factor, and a cam is cut for the machine that will automatically vary the power over the frequency range to give the same type power spectrum as shown in Figure 3, which incidentally, is far from constant power. This is especially desirable since it will eliminate the need to overdesign the equipment to meet imagined power levels at certain frequencies. By such a method an exact replica of the composite power spectrum encountered in practice (plus a safety factor) may be applied to the equipment. This is to be greatly preferred over any conventional improvised vibration test specification.

It is well to emphasize here that the power of the vibration given by Equation (2) is the specific power, the watts per pound of specimen. It is wattless and draws no power from the line regardless of the weight. The total line power required in the steady state of vibration is only that necessary to overcome the loss in the machine and to supply that absorbed in the specimen due to any internal deformation or damping. This latter power is the margin of real power which destroys the specimen if either deformation or damping is taking place. Generally it is too small to destroy or seriously affect the test specimen if acting along thereon (i.e., if there were no wattless power present). A further loss of power will result from any design of machine or driver which is incapable of storing power from the line or driver, such as with some solenoid types of drive. On the other hand, an important type of failure which is rarely accompanied by increased power loss is merely that of malfunctioning. Above some critical vibration power, a gyroscope will begin to drift intolerably, a clock will run off time, the power of a radio device may become undesirably modulated, or a vacuum tube may short. None of these considerations changes the fact that the vibration power impressed on the specimen is the same if the excursion is kept the same. This may, with some types of machine, require putting the vibration pickup on the significant part of the specimen rather than on the vibrating table, to be certain that the vibration power is maintained thereon.

By rearrangement of Equation (2),

$$E = 13.15\ P^{1/2}F^{-3/2} \qquad (2a)$$

$$F = 5.57\ P^{1/3}E^{-2/3} \qquad (2b)$$

But from Equation (1)

$$F = 4.41\ G^{1/2}\ E^{-1/2} \qquad (1a)$$

$$E = 19.45\ GF^{-2}. \qquad (1b)$$

Then

$$G = 0.676\ P^{1/2}F^{1/2} \qquad (3)$$

and

$$P = 2.19\ G^2F\text{-}1 \qquad (3a)$$

$$F = 2.19\ G^2P\text{-}1 \qquad (3b)$$

Also,

$$P = 0.497\ E^{1/2}G^{3/2} \qquad (4)$$

$$E = 4.06\ P^2G^{-3} \qquad (4a)$$

$$G = 1.597\ P^{2/3}E^{-1/3} \qquad (4b)$$

With the aid of these relations, it is possible to note more readily how a change in any variable affects any other.

Figure 4, illustrating these equations, consists of a "Siamese-twin" log-log graph, jointed on the frequency base line and coordinating on the "G"-lines. The outside left scales of power and excursion are worked against their corresponding solid G-lines, and the frequency line. The following example shows how the power may be found from the excursion. Enter the left excursion scale at 0.015 in. and move right, across this line to the solid 20-g line. Running down from this intersection to the center, the frequency is found to be 161 cps, Continuing down along the 161-cps line to intersection with the lower solid 20-g line, and turning left from this intersection to the outside left power scale, the value of 5.5 watts/lb is found. A quick slide rule check gives 161 cps and 5.44 watts/lb.

The power graph is extended across the center into the excursion graph by crossing the power scale over into the inside edge of the graph at 100 watts, and extending it to the top end with short markers, slant numbered, up to 10^7 watts. In the same manner the excursion graph is extended by crossing it over at 10 in. into the graph, to 10^6 in. This greatly increases the range and general usefulness. However, care must be used to work the left outside scales against only solid slant g-lines, and the inside left scales against only dashed slant G-lines. Given an extreme excursion of say 250 in. at 5 cps, the acceleration and power are found as follows: The excursion of 250 in. is found on the inside left scale of markers just below the 100 in. marker at 2.5. Interpolation is necessary here. Moving right, to the 5-cycle line, the intersection with the dashed excursion G-line is at 300 g's. Following the 5-cycle line upward to its intersection, by interpolation, with the dashed power 300-g line, gives 40,000 watts/lb on the inside left power scale, as a first approximation. A slide rule check gives 321.3 g's and 45.2 kw/lb.

To keep the graph more readable it was ruled only to a limited extent. It may be ruled to a finer degree by the user in the regions most used, to reduce the amount of interpolation necessary. To simplify the process of reading power from excursion and frequency, Figure 5 is given.

A further advantage in expressing vibration magnitude in terms of vibration power is that it coincides with the method of expressing sound magnitude in terms of sound power. Equation (2) may be reduced to db above or below any of the commonly accepted db levels; namely, the milliwatt (dbm), the watt (dbw), or the kilowatt (dbk). Hence, when equipment is tested with a sound power of say, 175 db, the air particle displacement (excursion) is known to be 0.489 in. for a

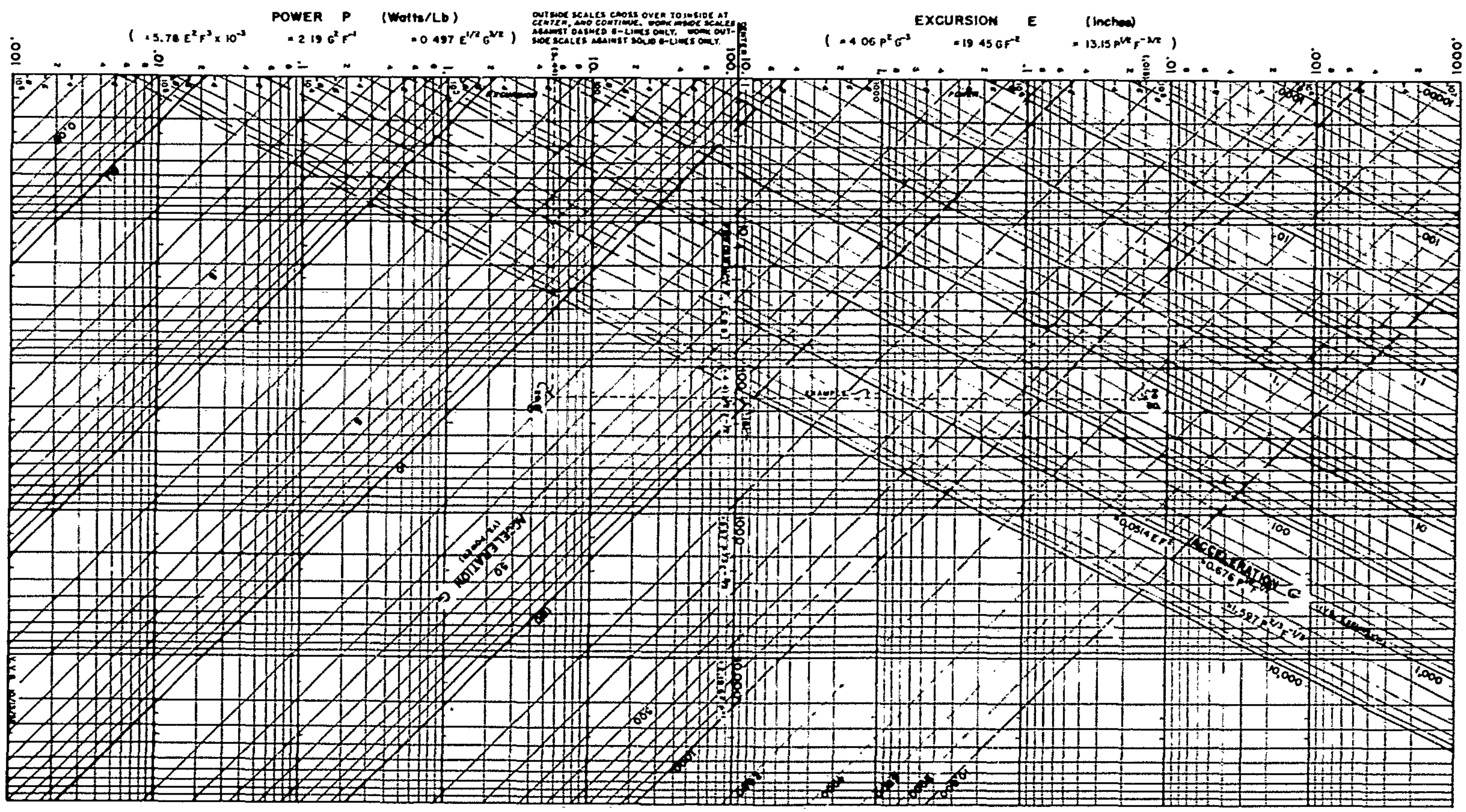

Figure 4 - Relationships of G, P, E, and F

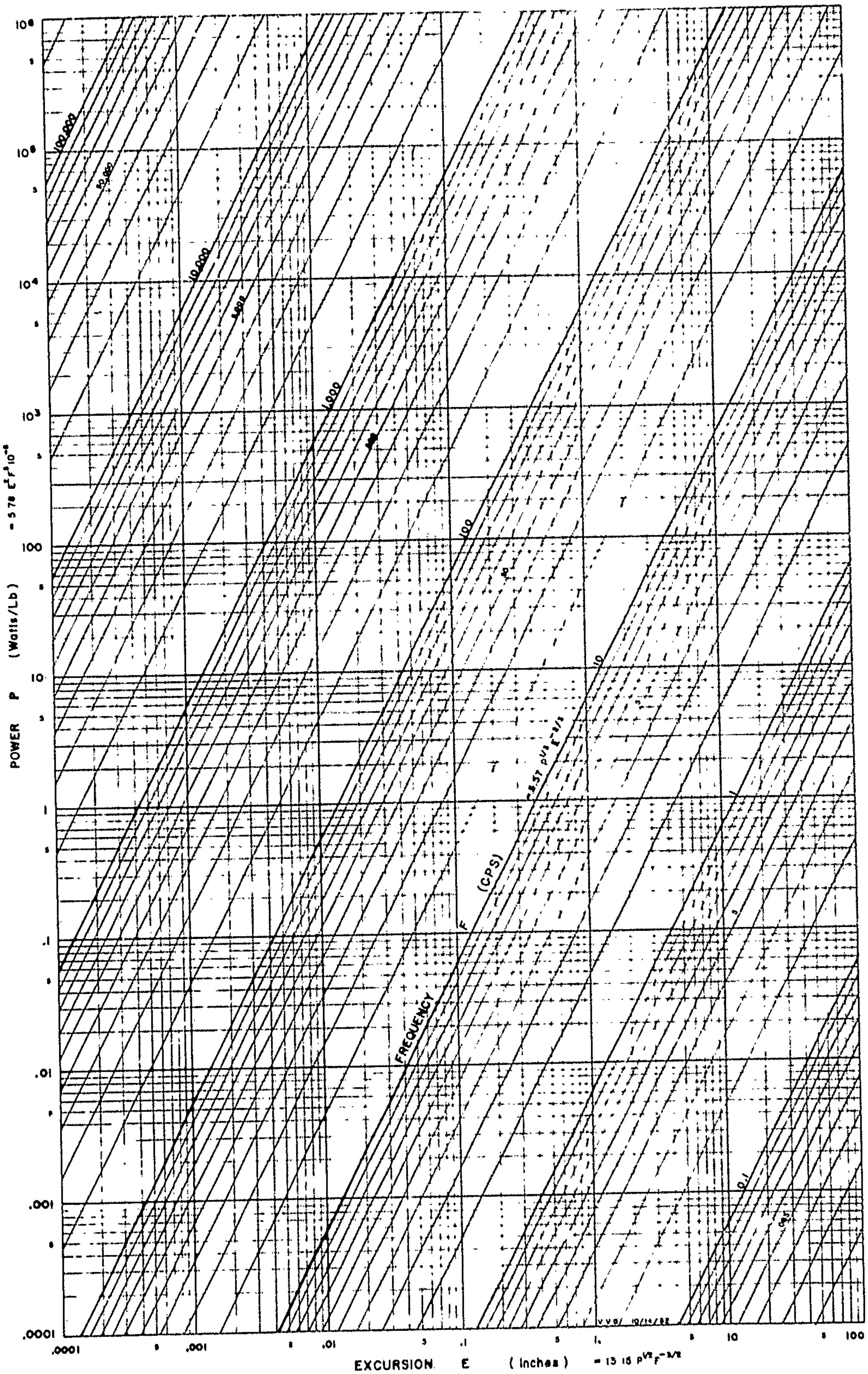

Figure 5 - Relationships of P, E, and F

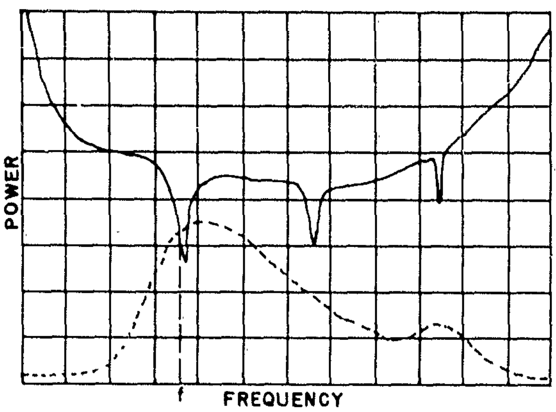

Figure 6 - Failure spectrum of an equipment (solid line) showing the malfunctioning power or breakdown power over a range of frequencies. The dotted line is the vibration-power spectrum for an aircraft (see Figure 3). Intersection indicates likelihood of failure at frequency f

tone of 1000 cps. (This represents an rms sound pressure of 114,700 dynes/sq cm = 1.66 lb/sq in.) On a power basis, by Equation (2), this is

$$5.78 \cdot 0.489^2 \cdot 1000^3 \cdot 10^{-3}$$

$$= 1{,}384{,}000 \text{ watts/lb of air.}$$

Thus the setting up of a noise intensity of 175 db may be accompanied with some difficulty.

The rms sound power in watts per square inch impressed on a test specimen based on a 0-db level of 10^{-10} watt/cm^2 is, for 175 db, 31.6 watts/cm^2 = 416 watts/sq^2 in.

By either method of calculation, the extent of the strain produced in the test specimen is dependent only upon the resultant excursion produced, or upon the response of the test specimen to the impressed noise power or to the vibration table. While it may be desirable to make certain that equipment remains operable during and after exposure to a noise having an air-particle excursion of 0.489 in. or a pressure of 1.66 lb/sq in., the vibration power at which the specimen is driven may be but a small fraction of the impressed noise power. Hence, the noise test may be replaced with a vibration-power spectrum which definitely imposes an equivalent-to-noise amplitude of vibration on the test specimen. Further, the power method of specifying vibration tests could be set up to eliminate the need for the noise power test by merely superimposing the equivalent noise power spectrum on the regular vibration power spectrum; i.e., by cutting a cam which reproduces the composite power spectrum of both the vibration test and the ..'se test. Thereby a single specification and a single test run would suffice for the variegated test runs presently specified for a single specimen, and would require less effort and expense.

To obtain the equivalent-to-noise power spectrum it should not be necessary to set up a noise intensity of 175 db. A much lower noise intensity of say 100 db might be set up and the response of the specimen measured. From these data the motional impedance of the specimen may be calculated approximately. The equivalent-to-noise vibration power, i.e., the specimen's response power at 175 db, may then be computed to a practical exactitude, multiplied by the customary factor of safety, and then applied on the vibration machine.

The foregoing results may be of the greatest practical value to the equipment designer. If he has the facilities to vibration-test his samples to either the malfunctioning power or the breakdown power over a range of frequencies, a "failure spectrum" may be plotted as shown in Figure 6. This is virtually a curve of the characteristic endurance of the specimen. To determine whether his component design will be able to withstand the vibration encountered in actual installation the designer need only superimpose a transparency of the composite aircraft power spectrum such as Figure 3, over the failure spectrum, or draw it as shown dotted in Figure 6. If any part of the failure spectrum overlaps the aircraft vibration-power spectrum, as shown, the component may be expected to fail at the frequency of the intersection, f, and either a fix or a redesign of the specimen is indicated.

It is not expected that a sudden change-over to the system outlined here can be effected. It is recommended that for quite some time the present specifications be retained, but that the corresponding power be inserted parenthetically as is done in the present general specification. At the same time, however, an optional power-versus-frequency specification may be added in each case, in either graphical or tabular form, so as to further encourage the change-over by modification, or complete new design, of test machinery. While it may take many years for old test machinery to wear out and make it possible to discontinue the present form of specifications of vibration tests, the change can be made, nevertheless, with a minimum of economic stress on the industry.

The author wishes to thank Dr. Charles Harry, Research Division, BuAer, for his review of the power theory.

Comments on this paper and requests for further information should be sent to the author at the following address: Bureau of Aeronautics
Navy Dept., Rm. 1W82
Washington 25, D. C.

DISCUSSION

J. Muller, Sperry: The fact that vibration is not measured in power has been our headache for a long time. The major trouble is that we have had no instruments with which to measure power. Everyone is measuring what he is able to measure-velocity or displacement—and is using any other parameters he has available.

It may be easier to produce a certain standard amount of vibrational power for test purposes, but the basic question is how to correlate this with actual field tests. In the field, we should like to measure the power in the particular areas that are most sensitive to vibration. As yet, we are not able to do this, in spite of the fact that many activities are working on the problem.

Gunsolley: You need only avail yourself of any of the more modern methods of measuring acceleration, excursion or frequency. Then you will have all the information you require to get the power spectrum and the endurance spectrum of the vehicle on which your equipment will be installed. Then Figure 6 can be used to predict whether the equipment is of satisfactory design. The specification will be concerned mostly with the over-all endurance spectrum of the machine, whereas the design engineer will be mostly concerned with spot analyses of critical areas. You are a design engineer, and you should know that power is the thing you are after. As long as you do not become too obsessed with acceleration, excursion, frequency, or energy, and keep power in mind, you cannot go far astray. However, before you turn a specification over to a test engineer, you should convert your data to power and give him a power curve or a cam for his machine that will reproduce the power spectrum exactly. Then, all he has to do is put the cam on the machine and let it run over the frequency range automatically. He can then devote his entire attention to the behavior of the test specimen, on a simple "go no-go" basis.

J. Markowitz, Barry Corp.: If power is a measure of the ability of the equipment to stand up under vibration, we should certainly think in terms of power. The fact that we cannot measure the power directly is of little importance. You cannot measure the power of a motor directly. You measure the torque and speed and you calculate your power. If expressing the phenomena in terms of power gives us a better picture of the situation, and a better means for comparison of all specifications, I am for it.

K. Unholtz, MB Mfg.: When I first read your paper, I thought that we were talking of power in a very broad sense, but in listening to the discussion I realize that power is used only as an alternate to the familiar displacements and accelerations. Particularly if you emphasize that power is wattless, it shows that you are thinking in terms of forces in phase with accelerations, rather than of velocities. If you are simply shaking a mass, there is only wattless power. If the mass is spring-coupled to a framework, you may have to worry about damping in addition.

However, it is possible to get phase shifts between transmitted forces, and motions of your shake table that are not necessarily dependent upon just the dissipation in the damper. As you go through resonance, for example, you may put energy into a system in the form of kinetic or potential energy that is exhibited by build-up across a local spring. It seems to me that if you are really going to use this quantity of power, it should have some meaning in terms of its destructive ability as such. I do not believe that you can preset the power-versus-frequency curve by using a simple cam arrangement; in order to get the profile of a cam, you would have to have a mechanical impedance installed in your airplane, or a test setting that would drain power from your environment in the same way in which you expect to put power back into your specimen in the test laboratory.

If you look at power in that very broad sense, it begins to take on meaning. If you restrict it to only the wattless power that you can obtain mathematically from either acceleration or displacement, I think we are falling short of the true goal.

Gunsolley: If the wattful power cannot be calculated conveniently, at least it can be measured and statistically related to the wattless power, which generally can be calculated readily. If you have several degrees of freedom present, you

can apply the power equation separately to each degree of freedom and add the whole vectorially, bearing in mind that in any series arrangement, each driven member becomes a driving member to the one following. The wattful power may be represented by a horizontal vector. The wattless power, lagging in phase by 90°, can be a vertical vector. The resultant vector will represent the over-all power seen by the line. When the damping power is negligible, the apparent power is nearly equal to the wattless power. When the damping power is greatly in excess (e.g., with very inefficient vibration machines; in ultrasonic vibration where it is made high deliberately in order to produce physical or chemical transformations), the apparent power is nearly equal to the wattful power. In your example, the power of the secondary mass, driven by the primary mass through the damping, though out of phase with the power of the primary mass, may be added thereto by either vector addition or rms addition, as appropriate.

If the airborne equipment is being designed with a minimum safety factor, for use in a specific location on a specific aircraft, then that location should first be loaded with an equivalent dummy load. A special power spectrum of the location should be taken so that when that same spectrum,, plus the required factor of safety, is put on the vibration table, the operational vibration conditions will be safely duplicated. Other than this, I see no need for the installation of a special mechanical impedance between the cam and the vibration table.

R. N. Janeway, Chrysler: Just what is your physical concept here concerning vibration power and how does it tie in to the ability of material to withstand it? On the one hand you say this is wattless power, and on the other you read from your chart the number of watts represented.

Gunsolley: The power I am reading would not show on a wattmeter, so it is called "wattless." The term is frequently objected to as being ambigious. The term "reactive" is more classical, but less descripitive. The power is wattless if it is not expended. It takes the same size and strength of electrical machine to handle 100 kv-amp of wattless power as to handle 100 kw of wattful power. The magnetic field, the torque, and the I^2R loss are the same in either case.

Janeway: How do we relate this power to the strength, fatigue life, etc. of the material?

Gunsolley: The correlation calls for a special study on each different type of test specimen. From the standpoint of the specifications, this is unnecessary. There, you merely wish to determine arbitrarily whether a given specimen will stand up under the operational conditions on the aircraft on which it is to be used. Until the day you discover the exact reasons for breakdown in any given type of specimen, the correlation for that specimen will, of necessity, remain statistical. In the meantime, I think you are quite safe in assuming that it is the power that is causing the breakdown.

Janeway: I still do not see the tie-up to the physical properties. In other words, can you interpret the power equation in terms of fatigue, for instance, which is, say stress times repetition?

Gunsolley: Fatigue is not considered in the derivation of the vibration power equation. Fatigue is not necessarily vibratory. It is something else entirely.

Janeway: But fatigue is necessarily present in alternating variable stress.

Gunsolley: That is true, but in general it makes no difference what the rate of alternation is. The S-N diagram is independent of frequency, except that a change in frequency may result in a change in stress.

Janeway: Well, higher frequency means that many more alternations in stress.

Gunsolley: It means merely that the higher the frequency, the more quickly you use up the number of stressings permitted by the S-N diagram.

Janeway: That may be where your power comes in.

Gunsolley: Perhaps. Higher power, when it gives higher stresses, means also a smaller number of stresses permitted by the S-N diagram. If, on the other hand, higher power gives higher frequency, it means that the permissible number of stressings are expended that much sooner. Fatigue is a work function, and certainly the higher the power, the sooner it will get the work done. So, you are likely right. Not all failures, however, are due to fatigue.

Frank C. Smith, NBS. There are three current theories on why materials fail. One is the maximum stress theory; one is the maximum strain theory; and there is one related to maximum strain, or energy. For any of us to say that this method is the best method or the most successful one, means that he must have some pretty good ideas on why materials fail. I would like to

know if any work is being done in the field of inspecting vacuum tubes and other components, for example, and if these things fail when subjected to a constant amount of stress of displacement or power.

Gunsolley: The theory of specimen breakdown is another subject, for many more papers. Failure may be due to any form of overload that causes overheating, deformation, collapse, or simple malfunction.

Markowitz: We are not trying to prove what the material will do under a certain power. I think the value of this power concept is mainly in making your laboratory tests a good equivalent of what happens in the field; and that is where we have greatly fallen down.

H. M. Forkois, NRL: I agree with Mr. Gunsolley that there are all kinds of power just as there are all kinds of shock motion. Some structures will not respond to certain types of power. If you have a specimen on a vibration isolator and you apply power to it at a very high frequency, the specimen will never see this power.

Gunsolley: If you put that same specimen on an aircraft that generates that same high frequency it will not see that power there, either.

Capt. J. E. Arnoult, WADC: I think the power effect on items of equipment is important, but I see no reason to assume that on an aircraft structure, for instance, the power is going to be constant throughout the frequency range of vibration. As a matter of fact, the data do not bear that out. As far as defining the testing of materials is concerned, we are primarily after an actual power curve.

Gunsolley: I did not intend to imply at any time that, in general, one would apply constant power over the entire frequency range.

C. E. Crede, Barry Corp.: This paper points out that the mechanics of damage resulting from vibration are not well understood, and that specifications often are confusing and inadequate as a result of lack of understanding of the principles involved. Most engineers would agree heartily with this comment, and would be in favor of any realignment of thinking that would tend to improve the conditions.

The contention of the author apparently is that an increase in the power embodied in the vibration test would bring about an increase in damage. This is perhaps true, but not basically because of the increase in power. If the power is increased, one or more of the displacement, acceleration, or velocity parameters must also be increased. This tends to cause greater displacements between components of the equipment and increased stress in structural elements of the equipment. In a general way, it may be stated that this increase in stress and increase in the accompanying strain is the cause of failure. The present inadequacies with regard to specifications result, in part, from lack of understanding of the relation between damage and conditions of stress and strain in the structures.

Figure 6 of the paper is quite interesting, but the following two qualifications should be pointed out in connection with this figure.

1. A curve similar to that shown could be plotted using displacement, velocity, or acceleration as a function of frequency, instead of power as a function of frequency. The same reasoning with regard to damage could be applied using these parameters and the same results could be obtained.

2. The upper curve apparently is drawn on the assumption that it is possible to determine a curve of vibration level, above which failure occurs. This is quite indefinite inasmuch as it does not take into consideration the number of hours required to cause failure. The fatigue effect tends to vary with the duration of the test. The usefulness of the method presented by the author is not apparent, nor is its validity proved. To some extent, the paper seems to contemplate the measurement of vibration power directly or, alternatively, the measurement of vibration in terms of amplitude with a subsequent conversion to power. It is difficult to see where this concept is useful, but neither is it incorrect if the influences of the method are kept in mind in applications of the data.

Gunsolley: As long as one keeps all the components in mind, they are mathematically an alternate for power, but power is the first cause. If breakdown acceleration is plotted against frequency, the statistical data will be in a convenient form, but it is misleading as a picture of anything but acceleration, unless it can be shown that breakdown is caused solely by acceleration. No one component of power can cause damage without the other components being present; hence, a power plot is the most useful for general comparisons and for general publication. The curve (Figure 6) may be based on any desired number of hours of fatigue. Obviously, a curve which gives the maximum power a specimen can endure

for 100 hr is considerably different from one which gives it endurance for 10 hr.

R.E. Blake, NRL: If I understand Mr. Gunsolley's proposal, it would make no change in what a vibration test machine would do to a piece of equipment. A specification based on a curve of power versus frequency would result in the same test as a plot of amplitude versus frequency. The only effect apparently would be a psychological one on the machine operator.

The emphasis on the term power seems to be based on an intuitive notion that the vibratory "wattless power" is the primary source of damage to equipment. It is of course true that an equipment cannot fail unless it receives energy, but the energy causing failure is that which causes a change in shape of the equipment under test. This may be called the strain energy. The equation of energy given by Mr. Gunsolley is correct only for a rigid mass—one which by definition cannot be strained or distorted. The energy of the equation is in fact the kinetic energy received and then given up by the vibrating mass during a cycle of vibration. The strain energy in the equipment is a correction term to be added to this equation.

It was suggested that the control of a vibration machine might be a better approximation to the desired value as a function of frequency if it were a control of power rather than amplitude or acceleration. However, the suggested cam controls amplitude of table vibration. Mr. Gunsolley appears to be objecting to stepwise approximations of specification curves. Certainly if these steps produce excessive deviations from ideal tests, a cam control is desirable.

Gunsolley: The operator is freed from the analytic mechanics of vibration, and is given a simplified program to follow. Thus, tremendous relief is provided. Since the runs are standardized by the machine, human error is eliminated. If I seem not to have considered the wattful power in my derivations, neither does the profession seem to have considered the wattless power in its derivations. My omission is further justified by the fact that the dissipative power is not inherently vibratory power, but is merely ordinary power modulated thereby.

Naturally, consideration of the wattless power does not change the effect of a present machine on a specimen. It is what the concept does to the improvisation of test specifications that is most important. Until power-calibrated machines become available, specifications must be given in both terms. However, one has nothing to lose if he finds himself faced with a power-calibrated machine.

Other comments—especially with regard to a nonrigid mass—are covered mainly in the replies to Mr. Unholtz and Mr. Janeway.

Dr. H. M. Trent, NRL: Methods of testing that will lead to smaller and less expensive machines are worthy of further study. An examination of Mr. Gunsolley's paper discloses a number of disturbing points. These will be taken up keeping in mind that parts of an apparatus probably fail because of excessive strains in the parts.

First, how does an investigator measure power in vibration testing? This is a tough question—one for which there is no adequate answer at the moment. It is my belief that the development of a good vibration wattmeter is highly desirable. Then, how can a specification be based upon power when it is not feasible to monitor this quantity?

Gunsolley: My answer is essentially the same as in my comments to Mr. Muller.

Trent: Someone may say that surely the electrical power delivered to a vibration machine can be measured and this should tell us something. To illustrate the fallacy of this remark, let us take an extreme but likely case. Suppose a device weighing 100 lb malfunctions in a vibratory environment because the grid of a tube goes into resonance. It is submitted that this resonance will never be disclosed in practice by measuring the electrical input power to a machine.

Gunsolley: I see little if any difficulty here. Suppose that we stop the machine at this resonant point, reduce the power setting, and put in a new vacuum tube. We then hold the frequency at resonance and keep increasing the power to a point where the tube shorts again. This point, then, is the bottom point on the first dip in Figure 6. The first tube failed at some point f in the same figure. The ratio of the power at f to that at the bottom point is proportional to the square of the amplification due to resonance. It is true that there may be little correlation between the vibration power and this type of failure. The tube shorts because of excessive amplitude of its elements, but the excessive amplitude would not be there if the power needed to produce it were not there also.

The fix to use may be one of many, after which the first dip will diminish or disappear. But before going to all the trouble and expense, it may be well to be sure that in practice the machine will ever be driven to the critical vibration power. The second dip is another such resonance point, but since the aircraft power spectrum falls far short of that point, no malfunction ever occurs, and the resonance is harmless. It would give trouble, however, if tested on an excessive power—based on imagination—and a fix would result in over-design. There is no necessity for the power to the machine to increase at the instant of failure. This would likely be a poor method of measurement anyway, unless the power increase were quite marked.

Trent: Mr. Gunsolley argues along this line: If we have a particle oscillating about a fixed point; if we know the maximum amount of energy stored in the particle in the form of kinetic energy; and if we say this energy is in some sense—not clearly defined—averaged over a quarter cycle, then we can compute a quantity $P = 5.78E^2F^3 10^{-3}$ watts/lb. Now P is related to the excursion and hence to the number of g's experienced by the particle. The particle does not break in practice; rather, it is something analogous to the spring connecting it to the fixed point. Gunsolley now places himself in the position that he cannot measure P, so he measures E (or maybe G) and computes P which he specifies. However, to determine if the elastic member breaks, he must recompute E.

Gunsolley: I am not aware that I averaged the energy at all; I merely integrated it. I obtained the classical kinetic energy expression $\frac{1}{2}MV^2$, in which, in this case, $V = r\omega$. Here, the frequency F is whatever kind of frequency the particle may have, resonant or otherwise. The power equation applies equally to forced and to free oscillations. I averaged the power, however, by dividing by the time of a quarter cycle. Hence the power is 4F times the energy. There is no objection to computing the excursion from the power to find the cause of failure, if one thinks that is it. The value of E should be readable directly from an automatic machine, simultaneously with the reading of power, if no resonance is involved. If there is resonance, the indicated value of E must be multiplied by the amplification of resonance; this will also mean a higher value of power than was indicated by the square of the amplification factor.

Trent: The process of basing an argument on a system consisting of a particle connected to a fixed point by an elastic spring, causes me some concern. Elastic members attached to nonmoving, rigid inertial points seldom if ever exist in actual vibration problems.

Gunsolley: I studiously avoided resonance and spring-connected masses in my paper, so as not to cloud the issue. My very first analysis (not shown in the paper) consisted of a flywheel driving a mass through a slotted or Scotch yoke, to produce a true harmonic motion, as shown in Figure 1. Such a system is nondiscriminative to frequency and, if ideal, will continue to run indefinitely at whatever speed it has at the time it is abandoned. No potential energy is involved, as in the case of resonance. The kinetic energy merely surges back and forth between the oscillating mass and the flywheel, without transformation. Of course, if the bearing of the flywheel is not rigidly fixed, then it may not produce a simple harmonic motion. Any rigorous solution by the power equation, then, must be preceded by a harmonic analysis, and the equation applied separately to each harmonic.

Trent: In describing Figure 3, Mr. Gunsolley states, "This does not give power as a function of frequency but power at any given frequency in the sense that no operations need be performed on the curve to get its meaning." (The underlining is the author's.) However, Figure 3 is obtained from Figure 2 by squaring each ordinate and multiplying by the cube of the abscissa. Then, to understand Figure 3, we must also understand Figure 2, for certainly we cannot generate new information by performing these operations. But what is the meaning of Figure 2?

Gunsolley: The paper was a first rough draft. It was so obvious to me that Figure 2 was statistical—and not a function of frequency, except in the most remote sense—that I overlooked stressing the important difference. It is obvious that Figure 3 is statistical also, and for the same reasons so remotely a function of frequency as to be hardly worth mentioning. I submit that something new has been added to Figure 3; the power equation, for one thing, and the resulting increased high-frequency response, for another.

* * *

DESIGN OBJECTIVES FOR PACKAGE SHOCK RECORDING INSTRUMENTATION

D. R. Craig and H. S. Youngs, Reed Research

The design objectives for shock recording instrumentation are the product of two major items: the purposes of the instrumentation and the limitations in sensing, recording, and interpreting the data. Each of these items is discussed in detail, and design objectives are described for instrumentatiu., to serve a few specific purposes.

Reed Research, Inc. has a contract with the Packaging Development Branch of the Engineer Research and Development Laboratories to design an instrument for measuring and recording the shocks received by a simulated package during actual shipment. This paper reports a very early stage of the project - consideration of the design objectives for such a recording device.

The purpose of the recorder is to supply data to a packaging engineer that will enable prediction of the treatment a real package will receive when shipped. The environment to be assessed is that of real, and not simulated shipment. The recording period is to be long enough to permit transcontinental shipment and return by common carrier. The recording unit itself, built into its shipping container, must look, feel, sound and smell like any ordinary package and be of such size that it can be handled by a man. It must not only tolerate, but must also record shocks that are capable of damaging normal shipped items. The device must be inexpensive enough to allow use of many such units.

Thus, there are four qualifications - one major and three minor:

(1) The recorder must be able to accumulate meaningful and useful shock data;

(2) It must simulate normal packages;

(3) It must be reliable under extreme conditions; and

(4) It must be cheap.

These four qualifications may also be considered generalized design objectives, but only in the sense of idealized goals. Specific design objectives for devices of this type are the product of such purposes and of the limitations of the equipment; they are statements of realizable and not ideal purposes. In order to proceed from general to specific objectives, we must first consider the various functions to be performed by the instrument.

Our major purpose is to supply meaningful and useful shock data. "Meaningful" and "useful" are terms that involve the intended use. "Meaningful" refers to the input side of the instrument—a measure of the adequacy of description of the damaging qualities of shocks that can be identified by the device. "Useful" is on the output side—a measure of whether the data are in a form that serves the needs of the packaging engineer. We now have a description of a traditional "black box," whose input is the series of motions imparted to a package in shipment. Of these motions, it senses those which represent potential damage to shipped

items, records them during the trip, and finally provides an output to the investigator that enables assessment of the shipment environment for that trip.

Working forward from the output, we can begin to describe the requirements for our black box. First, the assessment of the data will almost certainly be statistical. The investigator has control over too few variables during the shipment for any other approach at this time. The immediate purpose of the investigation will be to compare the effects in terms of statistics derived from the trip with the causes, in terms of the description of the package, its origin and destination, its mode of travel, and the shipping instructions. These latter items are his sole means of control, and their use requires parallel shipment of a large number of similar instruments. Statistical assessment requires storage of data either in numerical form or in a form that can be converted to numerical without excessive effort.

The occurrences under investigation are those of damaging or potentially damaging events. The data that are reported correspond individually to aspects of individual shocks. One very important point is that the literature does not show what aspects of the shocks really have damage-predicting capacity (References 1 and 2). Statistical reports from shock recorders in transit are usually restricted to summaries of the probability distributions of accelerations as functions of frequency or of gross peak accelerations as a function of acceleration magnitude. Detailed reports are usually restricted to the detailed velocity- or acceleration-time curves for particular shocks. Shock damage reports have been limited to construction details and mechanisms of failure and prevention, both in field reports and reports from the use of testing devices (References 1-5).

In other words, we can characterize shocks readily by numerical means—summation of acceleration peaks and frequency components and the like—but we do not know that we are properly characterizing damaging or potentially damaging features of shocks. A useful addition to the data would be a direct indication of shock damage together with a detailed record of what happened just prior to and during the damaging events. This type of record could be regarded as a means of standardizing the statistical data, or as converting probability distributions of shock events to probability distributions of damaging shock events. To do so requires a time delay (or a memory) in the recording system, and a means of identifying damage, presumably an elusive problem.

Let us consider the memory within the device. The digital computer field is instructive in functionally describing a memory section in terms of the number of binary digits that may be stored per unit volume of recording medium (Reference 6). If our output is to be numerical data or data readily converted into numerical form, computer memory mechanisms will include powered systems that are satisfactory for our purposes. Appendix I gives some data with respect to these systems.

Unpowered memory systems are of interest in the instrumentation, if for no other reason than reduction in instrument volume. In this instance, we are using the shock itself as the energy source for making the record. This method is a valid way to record peak values or values that are summations over a whole shock. It is not particularly valid for use in a device that utilizes a delay system, to record initial aspects of a shock that is later classed as worth recording by another component of the instrument. An unpowered system is also limited, in that amplification is not available to assist sorting and refinement of the recorded measurements.

In general, a power-consuming, memory device, with all of these functions of the recording medium and mechanism, can be regarded as limited only in volume, cost and fragility. The volume and cost for a given medium can be considered roughly proportional to the maximum possible number of elemental records. An elemental record is defined timewise by the allowable time error, and numerically by the inverse of the fractional accuracy (i.e., if errors of ± 0.01 sec and five percent in amplitude are allowable for each variable in a 4-channel continuous record of 100-hr duration, $4 \times [(100 \times 3600)/(0.02)] \times (100/5) = 1.4 \times 10^8$ elemental records must be supplied). The space on the recording medium occupied by each of these elemental records is determined by the writing and reading system. Power-consuming memory systems do not carry additional inherent accuracies or inaccuracies aside from those due to fragility.

Memory devices not using power carry an additional limit of accuracy due to their energy demands from recorded shocks. Ability to record a variable with this type of system depends upon availability of a suitable mechanical linkage for isolation of the desired variable.

Sensing can be regarded as a filtering procedure; only the items desired are passed into the recording system, and the rest are ignored. Powered sensing systems enable amplification and thus permit some increase in ruggedness by decrease in the necessary mechanical responses of the sensing devices. The basic data available in the environment are linear acceleration, angular velocity, and acceleration variations with time. An item to be recorded will be a function of one or more of these. Wherever possible, it is desirable to use the concerned basic variable rather than one of the others and incur a differentiation or integration error. Matching of sensing and recording units is necessary, and possibly conversion from smooth to digital data between the two units. The most well-developed sensing and recording units utilize conversion of mechanical movements into more versatile electrical form. A large number of these units are listed in References 2 and 7.

The above comments allow us to state our design objectives, with the realization that we are not preparing design objectives for one, but for two instruments: A shock damage standardizing device and a statistical shock data collector.

SHOCK DAMAGE STANDARDIZING INSTRUMENT

This instrument shall perform the following functions:

(1) Record accurately and intermittently, full and continuous package behavior as a function of time for intervals characterized as damaging by some frangible component or components. These intervals shall be long enough to include the whole sequence of events that would be lumped together as "a shock", and should include the initiation period prior to the identifying breakage. A recording time delay is required, and hence a powered system.

(2) Produce a record that can easily be read and standardized.

(3) Sense and record either vector sums or three orthogonal components for linear accelerations. Comparable recording of angular quantities is desirable.

STATISTICAL DATA COLLECTOR

The collector shall fulfill the following requirements:

(1) It shall record at minimum allowable accuracy the variables found to be predictors of shock damage. The record shall be either numerical and summational or shall be capable of insertion in a high-speed reading device for such data reduction.

(2) It shall be adaptable to production of essentially identical units.

BOTH TYPES

Both instruments shall have the properties listed below:

(1) Basic dimensions and weights that will allow transport and handling by all normal common carriers: Say 1 cu ft volume and 50 lb maximum weight;

(2) Adaptability to larger packages;

(3) Low cost;

(4) Ability to sense and record shocks capable of destruction or serious damage to contents of normal packages of like description;

(5) Type of construction that permits simulation of normal and uninstrumented packages of similar description;

(6) Ability to operate untended for a 30-day period under normally expected extremes of temperature, pressure, and humidity.

In addition to these design requirements, a few items are worth a little discussion. The first of these is in respect to the variables that can be measured by a device within a shipping container. Obviously a shock can be interpreted in terms of energy or momentum transfer, of impulse delivered, of acceleration, velocity or displacement curves as functions of time, or of almost any combination of the fundamental units of mass, length and time. If any of these directly measure the damaging characteristics of shocks, we may say that a frangible element in turn is a direct measure of the concerned variable. At the present time, however, we only have the possibility of comparing breakage with physical variables, and must therefore concentrate our efforts towards measuring primary or elementary variables with the intent to calculate potential or plausible shock parameters from our data.

Within the shipped package, our only measurable linear primary variable is acceleration as a function of time. No external coordinates penetrate our container to allow direct velocity or position measurement. Angular positions, velocities and accelerations are all directly measurable because of the gravitational reference frame. These plus time and measurable breakage constitute our primary variables for the shock damage standardizer.

The next item of interest is the output of the statistical data collector. The statistics that are worth recording for a packaging engineer have two important limitations. First, they must characterize the potential damage that a shipped package could suffer; and second, they must describe features of package exposure where the packaging engineer can take protective and preventive measures. Other statistics are more or less beside the point. Characterization of damage calls for careful examination of individual detailed shock records that are identifiably damaging. It is probable that more than one variable is involved. Factor analysis can be a helpful tool in identification of these variables. It is briefly discussed in Appendix II.

It was mentioned that the data must be stored in either numerical form or in a form that can easily be converted to numerical. No single record will stand alone, if for no other reason than the existence of the controllable variables in the shipment of the particular package. For reliability, a large number of shipments will have to be made and the records examined. Let us say that something on the order of 100 thirty-day records are the basis for an analysis. If the record for each shock must be examined by the investigator, he will be examining records at about the same rate as they are produced, and his analytic base will require about 10 man-years of examination. The proper numerical form is summational—the kind of thing that will let the investigator establish his base in about 1 percent of this time. It is possible for this to be achieved on an individual shock record by a computer. Somewhere in the system a vast data reduction must be accomplished through the use of automatic mechanisms.

An internally powered system is probably unavoidable. This power will be needed for adequate control and operation of the recording part of the unit, and probably for the linkage between the sensing and recording components as well. The ideal objectives of "look, feel, sound, and smell like an ordinary package" may remain ideal without practical attainment, especially "sound." Faint clicks and hums will be almost impossible to eradicate in a reasonably inexpensive powered device. A practical objective would be to have any sounds, feels and smells as inconspicuous as possible, let us say inconspicuous to an individual carrying the package in his arms, where he is normally inattentive. Perhaps the tick of a good wristwatch is an allowable and comparable maximum.

Electrical powering is desirable in comparison with mechanical powering on the grounds of greater versatility, higher energy concentration per unit volume, ease and convenience of use and control, and availability of reliable amplification procedures and sensing units. Our restrictions do not prevent the use of electronic systems in amplification and recording, because they may be carefully protected by shock mounting and the like within the package.

APPENDIX I

DISCUSSION OF MEMORY SYSTEMS

Let us consider a continuous record drawn on a paper tape by a pen that moves laterally to indicate the magnitude of the variable. Let us say we wish to record to ±5 percent accuracy up to 100 cps. If we have a 1/2-in. total swing to the pen, at any instant the pen should be within ±5 percent of 1/2 in from its ideal position, or should be somewhere in a 0.05-in. lateral range centered on the real value of the recorded variable. The pen also should be within a lengthwise distance that corresponds to about ±5 percent of the time for one cycle at the limiting 100 cps frequency—about 1 ms.

Let us consider our 1/2-in. tape to be covered with an orderly graph-paper-like array of 0.05-in. squares. Our continuous record can be interpreted as an identification of a series of these squares along the tape, with one point per square. Since there are 9×10^4 sec per day, our record will be $9 \times 10^7 \times 0.05$ in. long. If we are using tape that is 0.004 in. thick and 0.5 in.

wide, this means we will need 5 cu ft of tape for a 24-hr continuous record, and that we are recording 9×10^7 measurements in it. The total number of possible elementary measurements in this volume is 9×10^8 (total number of squares), or 2×10^8 per cu ft.

Five cubic feet per day is rather discouraging. Let us think about photographic film as a recording medium, where we do not oscillate the marking beam of light, but instead vary its brightness. With this system, we can attain our desired ±5 percent accuracy. Let us further provide a mechanical system to use completely the film by re-running it in an adjacent channel when we have finished marking one line on all the film in the reel. Per Reference 6, we can put 50 adjacent channels on 35-mm photographic film and can mark 10^9 elementary data spots (comparable to our graph paper squares) per cubic foot of medium. One day's record, as before, will use 9×10^7 of these spots, or will require about 0.1 cu ft of film (about $100 worth).

These two examples illustrate the three features of a recording medium that are available for identifying and measuring a variable and for determining the time interval represented by a minimum elementary record: Two dimensions of record surface plus an additional dimension of record intensity. Our paper tape example did not use intensity variation and had to make up for it by loss of available record surface. In general, for any record medium, the basic elementary data can be considered to be attached to a minimum record surface element whose dimensions are established primarily by the recording and reading equipment. A good practical limit for the smallest size of these recording areas is supplied by Reference 6, in a tabulation of the maximum binary digit capacities of high-speed computer memory systems. These capacities are expressed in terms of binary digits per cubic foot of medium, and correspond to the total number of elementary squares in our paper tape example. These capacities will not be changed appreciably when the media are used to represent variable measurements by intensity variations. Table 1 lists some of these digital capacities with their approximate costs per cubic foot of record. Practical limits to the maximum number of identifiable intensity levels in an elementary data area are also given. The capacity and cost data are from Reference 6, except for our paper tape example.

Of these media, it may be seen that intensity variations cannot be used for accuracies much higher than ±5 percent tolerance without trickery involving more than one channel per variable—and such trickery is volume-consuming. Use of intensity levels as a means of identifying variables is also volume-consuming, as is use of telegraph-like multiplexing techniques. Therefore, if accuracies higher than within ±5 percent are to be desired, incomplete use of package or record volume is to be expected. If accuracies within ±10 percent or poorer can be tolerated, magnetic intensity variations become practical, and the available capacity of photographic techniques is not used.

Let us consider a 30-day continuous record of a single variable to an accuracy of ±5 percent variation, where 100 cps is the maximum frequency of the variable that is to be recorded to the limiting accuracy. The number of elemental measurements is 2.6×10^9, the number of milliseconds in 30 days. Each measurement

TABLE 1
Approximate Digital Capacity, Cost, and Intensity Data for Several Recording Media

Recording Medium	Number of Binary Digits/cu ft	Cost of Medium ($/cu ft)	Approximate Maximum Number of Intensity Levels*
Teletype punched tape	3×10^7	10	1
35-mm photographic film	10^9	10^3	10
1/4-in. magnetic tape	3×10^9	3×10^2	5
0.004-in. diameter magnetic wire	10^{10}	4×10^3	5
Ink-marked tape	2×10^8	10	1

*These are not theoretical, but practical maxima for untended devices. Under very carefully controlled conditions (References 8 and 9) photographic systems can attain perhaps 1 percent accuracy and magnetic systems can approach this figure. Such accuracies require too much refinement for our purposes.

is a choice among 10 alternatives, or the record medium must contain 2.6×10^{10} possible alternative elemental measurements. These will be in 2.6×10^{9} elemental recording areas, if 10 or more intensity levels can be distinguished in the medium; or in 2.6×10^{10} elemental recording areas, if less than 10 intensity levels can be distinguished in the medium. The corresponding number of elemental measurements and of possible alternative elemental measurements for the same recording period and maximum frequency, and for within ±10 percent accuracy, are 1.3×10^{9} and 6.5×10^{9}, respectively. Table 2 estimates the cost in terms of record volume and record medium expense for both of these accuracies and all of the media of Table 1 except the punched tape which consumes too much power. For other limiting frequencies at these accuracies, the volume and medium expenses are proportional to the frequency.

If a 30-day recording period is desired for a large number of identical packages with many variables each, even ±10 percent of accuracy is not particularly attractive. If we assume a 1:1:1 ratio of volume required for power to record to sensing plus recording plus controls, we are just barely able to record one variable continuously for 30 days to 10 percent of accuracy and 100 cps maximum frequency in a 1-cu ft package using magnetic systems.

The above discussion suggests that an objective is to provide instrumentation capable of pre-recording data reduction, to enable relatively long term recording of several variables. Progressively, we have a whole series of alternative means, ranging from intermittent operation of a recording unit that makes a continuous record, through devices that characterize individual shocks by one or more numerical indices, to devices that record totals or integrals or some other general feature of a whole trip. The important question is what kind of data reduction we require for adequate description of the kinds of variables we wish to record.

TABLE 2

Approximate Record Volume and Cost for a 30-Day Continuous Record of One Variable to 100 Cps and within ±5 Percent and ±10 Percent Accuracy

Recording Medium	±5 Percent of Accuracy			±10 Percent of Accuracy		
	Number of Elementary Areas Used	Volume (cu ft)	Cost* ($)	Number of Elementary Areas Used	Volume* (cu ft)	Cost ($)
35-mm photographic film	3×10^{9}	3	$3000	10^{9}	1	$1000
1/4-in. magnetic tape	**3×10^{10}	10	3000	10^{9}	0.3	100
0.004-in. diameter magnetic wire	**3×10^{10}	3	12,000	10^{9}	0.1	400
Ink-marked tape	**3×10^{10}	150	1500	**7×10^{9}	40	400

*The recording media are assumed to be used as efficiently as possible.
**Intensity variation is not available for use in recording.

APPENDIX II

MULTIPLE FACTOR ANALYSIS AS AN AID IN SHOCK DAMAGE INVESTIGATIONS

The lack of agreement about causation and adequacy of the several measurable parameters of shocks as predictors of shock damage indicate that more than one variable is involved. In a situation of this type it is desirable to use an analytical method that will identify the related variables and if possible show the mechanism of relationship. The number of details of transportation and handling and of the nature and complexity of shipped items impels the use of a statistical means of assessment. There are a series of statistical procedures for handling this class of problem, one of which is described in the following quotation from the summary given by Reference 10 and described in detail in Reference 11. "In the field of a 'pure science' it is possible to plan an experiment or series of experiments to confirm or refute a particular hypothesis. It is possible to construct experimental systems where all but one variable, or at most, a very few variables, are held constant. Interpretation is a matter of comparing the results with the calculated effects. This class of procedure implies complete knowledge of interrelations between the experimental variables and a high order of control of all the experimental conditions.

"The above implications are not satisfied by a wide range of sciences. In some cases, the

number of variables is too large to enable one-by-one manipulation to be used as a means of experimental procedure. It would require too long a time and would be too expensive for justification. In some other cases, the theoretical background is not secure enough to enable certain calculation. In such cases, auxiliary logical tools are required to overcome the limitations of the traditional experimental procedure.

"These auxiliary logical tools may be subjective, based upon long experience in the field, or objective, and may involve procedures for manipulation of the experimental results to attain intelligibility. In either case, the major questions that must be answered for use of such an auxiliary tool are 'is it efficient in its use of the available data?' and 'does it imply more accuracy than it actually possesses?'

"The form of multiple factor analysis presented in this Article is an objective auxiliary logical tool. Since it is statistical in nature, its efficiency in use of data can be calculated. As a matter of interest, it cannot compete with the efficiency of the traditional interpretive procedures until the number of variables concerned are large in number (let us say, on the order of 10 or more). With respect to implication of accuracy, the errors in its use are calculable, and in technical areas of this type, its inaccuracies are limited in effect, since its use is only to indicate avenues of profitable approach by other methods.

"The over-all procedure is to convert the mass of experimental data into statistical numbers that represent the degree of 'relatedness' of all pairs of variables. These numbers are interpreted geometrically and are, in turn, manipulated to achieve a geometric picture of the 'relatedness' of the variables in groups. The original data are then re-used to disclose the class of relationship that is involved in these groups of variables. Traditional methods, such as graphical procedures, are convenient for the purpose.

"The logic of this form of multiple factor analysis may be explained by reference to a hypothetical series of experiments. Let us visualize a mass of experimental data arranged in tabular form. Let there be N experiments represented in the table, a row per experiment, with the results of each experiment expressed as measurements of n variables, so our table will have n columns, one per variable. If each of these variables is completely independent, we may construct a Cartesian coordinate system of n dimensions, where each variable is assigned to an axis. We may then represent the table of data by N points in this n-dimensional space, where each point is defined by its $\underline{n}$ projections on the coordinate axes. The table of data and the n-dimensional plot are completely equivalent.

"Now if one of these variables, let us say variable $\underline{a}$, is really a function of some other variables, let us say $\underline{b}$, $\underline{c}$, and $\underline{d}$, then except as a check on experimental errors, column $\underline{a}$ values may be calculated from the $\underline{b}$, $\underline{c}$, and $\underline{d}$ values. Therefore, we may express the set of experiments in n-1 columns, or the corresponding plot in n-1 dimensions. Thus, interdependence of the variables is reflected in a decrease in the number of variables or dimensions required for expression of the experimental results, where those experimental results are expressed only to the accuracy implied by the errors in measurement.

"Let us assume that we have such a table and plot, and that we have $\underline{f}$ columns and axes, respectively, to represent the experimental data, where $\underline{f}$ is equal to or less than n. If we were to rotate one or more of the axes to a new position, and leave the N data points untouched, we would still have an accurate description of the experiments. The relative positions of the points rather than a particular set of numerical values represents the information gained by the experiments. The same could be said of the new f-columned table, created from the old one by an operation equivalent to the rotation of axes.

"We may express each experiment in terms of the original $\underline{n}$ variables as well as by such an arbitrary set of mutually right angled axes. This follows from the original description of the $\underline{f}$ axes, since we may say that each of the $\underline{n}$ variables may be represented by one of the axes or by a line determined by two or more axes. Therefore, we should be able to draw a new set of $\underline{n}$ lines in the f-dimensional space, where each line represents one of the original variables. These lines would be fixed by the N data points, regardless of the orientation of the arbitrary coordinate axes used to describe them for a particular plot.

"Thurstone's procedure supplies a means of finding the relative positions of such a set of $\underline{n}$ 'variable-representing' lines in this f-dimensional space. He shows that the angular

separation of these lines is a function of the 'relatedness' of the variables concerned, and may be used to isolate groups of variables that are connected in the set of N experiments. With this group isolation, we may return to the original data for investigation of the cause of connectivity. It was from groups of variables isolated in this manner that the relations were found that were reported under 'Outline of Results.' "

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DISCUSSION

H. L. Stout, SCEL: You mentioned the instrument that we are developing for making tests of this nature. I believe that the instrument has been described before this symposium by Dr. Goodwin. We expect delivery of an experimental model in three or four months, and then we are going to get to work and get some data with it. We feel that we would like to describe a shock pulse by means of a peak value of acceleration, and duration.

After we have had some experience with the instrument, we hope to have some more to tell you.

F. Mintz, Armour Research: It seems to me that you pose two alternatives, one of which is too simple, and the other too complex. The first is trying to describe the shock by a single number. The second is trying to define the shock adequately; this requires too many numbers and a rather complex piece of equipment.

In the field of naval shock, a concept has been evolved which is called the shock spectrum. This concept may be used to define a shock so that data may be obtained in terms of the possible response of packaged or ship-mounted equipment. The shock spectrum concept is much simpler than, say, a time-history-of-acceleration curve.

The shock spectrum is a plot, versus frequency, of the maximum effective acceleration

experienced by a simple system of a given natural frequency. The way to obtain this plot is simply to use a reed gage. Much of the shock spectrum work originated at NRL. You can operate with the shock spectrum on the known parameters of a simple system and come out with some answers about the response of the system.

There are certain shortcomings of the reed gage that we all know about. First, there is its lack of sensitivity at high frequencies.

Secondly, the gage has not yet been developed to a stage where it can be used for the kind of program we are discussing. It seems to me, however, that its development would be very much easier than the development of the rather complex system you have been talking about. Also, you will need to operate on the data in a much less complicated manner in order to get out the information you want.

It seems to me that this approach should be explored more thoroughly in the near future to see if we can get the answer we want.

Craig: To what form would this data from the frequency spectrum be reduced? Would you not try to reduce it to one number so that you could eventually plot distribution of the number of shocks against the severity of a particular shock? This seems to me to be the only practical way to predict the chances of a container receiving potentially damaging treatment.

Mintz: I believe that the vibrating systems are too complex to be treated by one number. The simplest number you can use, if you want to speak in terms of digital presentation of data, is some summary of the respective accelerations over various frequencies. It has already been stated that trying to define a shock by one number such as acceleration can be highly misleading.

You mentioned the concept of impulse. As a matter of fact, you can treat the reed gage data so as to obtain data that corresponds to impulse. If you have an impulse, or a step change in velocity, the shock spectrum will be a straight line, starting with zero acceleration at the origin and increasing linearly with the frequency.

Now if you have had an ideal step change in velocity then you could express the shock spectrum with one number which could represent the slope of this straight line. This slope is a measure of the magnitude of the velocity step, and with a proper scale factor, could be expressed in terms of the velocity change. However, as few shocks can be ideally expressed in this way it is very improbable that a single number can often describe a shock adequately.

Craig: It seems to me if we find a single number which will express conditions to an accuracy of ±10 percent, we have made a large improvement over the situation where more than half of the equipment which was shipped arrived in an unuseable condition.

Mintz: I agree. Ten percent, as a matter of fact, would be a high order of accuracy for this problem.

Jesse Steinman, Hughes Aircraft: When you travel around the country, the one thing that impresses you is the number of different modes of transportation there are, particularly the number of different types of trucks and railroad cars. This does not include different modes of transportation for overseas shipment.

I wonder if either you or Mr. Ashley ever thought of how much time and how many instruments would be required to get the data that you consider necessary for designing for shock and vibration?

R. B. Ashley, NBS, Corona: Dealing in round numbers, let's say something in the neighborhood of a million miles, and about a hundred instruments. We can add zero to each of these if we like. The problem of damage occurring during transportation is an overlapping function of the military services. It was for this reason that I proposed that each of the services concerned with guided missiles should determine statistically the hazards to missile packages in their particular modes of transportation. This work could be done only after the proper instrumentation has been developed. All such data could be combined by a central agency into useful design information. The same instrumentation that proved to be desirable for obtaining the data under service conditions, could be used for calibrating of test instruments to see whether the test procedure and test equipment develop the same type of hazards as are encountered in the services. However, I think that more instruments shipped more ways will result in more accurate statistics.

Steinman: Have you made a time estimate on how soon the information would be available after completion of the instruments?

Ashley: Yes, the program would be complete when we stop shipping.

Craig: Which cannot be expressed as one number, incidentally.

H. M. Forkois, NRL: The small reed gage developed by Dr. Vigness is on display in Building 10. The instrument is very small and very compact, and is worth looking at.

H. L. Rich, DTMB: At the symposium you mentioned two years ago, I gave a paper describing the evaluation conducted at the David Taylor Model Basin of the then existing reed gage. I mentioned at the time that a new reed gage was being developed. This new instrument has been completed and has been in use during the past year. Two versions of the gage are on exhibit in Building 10. One of them is the standard gage used for ship shock work. It has a stationary record platen across which waxed paper is stretched and secured by rollers at each end. The paper does not move during the time interval required for recording. The second type on exhibit is one to which an electric motor has been attached. A continuous time record of the shock spectrum is obtained.

I realize that for packaging, the tape would have to be run very slowly in order to get a sufficiently long record with a reasonable length of paper. However, I see no reason why the drive mechanism could not be modified so that, first, a larger amount of paper could be carried, and second, the paper drive would operate only during periods in which the shock is above a particular level and is being registered.

Dr. M. G. Scherberg, WADC: Twice now this morning there has been mention of a need for nine peak reading accelerometers. I am rather curious about this recommendation.

R. L. McKay, USERDL: Peak accelerometers record some kind of parameters composed of peak acceleration and duration. This response has been analyzed for square waves and I have done it myself for sinusoidal pulses. It is a little hard but it can be done.

I wish to put in a plea for a reed recorder that will give a record of peak accelerations.

J. P. Walsh, NRL: I feel that it can be shown conclusively that a measurement of peak acceleration has little or no bearing on the problem of damage to multi-degree-of-freedom systems, which I assume is what your are packaging. I think Mr. Mintz's suggestion that the shock spectrum is the most meaningful parameter of a shock motion that we know, is perfectly sound. Simplicity is certainly to be desired, but one must be sure that he doesn't oversimplify.

Rich: I would like to confirm, from our experience, what Mr. Walsh has said. We have gone through the same sort of development. We started off originally by using peak reading accelerometers. We found that the data, first of all, was very unreliable and not useful.

Next, we developed contact-making accelerometers. The data obtained with these instruments, likewise, was simple to obtain but very difficult to use.

Then, we developed the reed gage. Now, the reed gage, which being no panacea for our problems, is certainly much closer to being our answer than any peak-reading accelerometer. I believe the same sort of development work is going on in the packaging industry. There is a tendency to use an instrument because it is simple, although simplicity in itself is not what is desired. We need useful information, and as Mintz and Walsh have said, it is going to be necessary to use somewhat more complicated instruments in order to get useful information for the design of either packages or of the equipment itself.

Percy Ott, USNOTS: I would like to mention one criterion of damage in the early studies of transportation shock by the American Association of Railroads. The criterion was the number of sides of beef found on the floor of a refrigerator car. We did find there was a very good correlation between the sides of beef down and the maximum shock that occurred on the ride. By "very good" I mean, roughly, a 50 percent correlation. When we got the level of peak acceleration down below, say 2 g's, the number of sides down was relatively small. Later, another criterion was established; it was the number of crates of cranberries which were scrapped.

This is the type of data we need—how many vacuum tubes get broken on a ride versus the number of something, whatever it is. I think that when we use peak accelerometers, we are aware of what we are measuring. We are really measuring impulse above a certain level and we can make the gage give us a stiff combination of peak times time; it is going to be rough, but it is better than nothing.

C. E. Crede, Barry Corp.: I would like to add my support to the comments of Mintz, Walsh,

and Rich in support of the reed gage. I believe that with regard to the sides of beef, if something is done to change the operating procedure which would cause more sides of beef to fall on the floor, there is a reasonable chance that perhaps the peak accelerometers would register smaller accelerations than before the change in procedure took place. However, persons faced with the problem of designing packaging for items to be shipped in transit, cannot obtain very much information from peak accelerometers about the strengths required in the equipment to be shipped or the stiffness of packaging needed.

If we are talking about a simple instrument, I believe that the reed gage to which Mr. Mintz referred is the most adequate simple instrument of a peak-reading type that we can possibly use. However, Mr. Rich's motorized reed gage seems to overcome one deficiency of the simpler instrument by giving information, not only on the responses of structural mechanisms of various natural frequencies, but on the number of recurrences of shock of various levels.

Jesse Markowitz, Barry Corp.: I agree that if you get enough accurate information, it will make sense statistically. However, if the information includes many variables that have not been classified, you have to accumulate more statistics to make it make sense. If you have no basis on which to compare the shipping damage by plane, train, bus, truck, and other forms of transportation, so many variables are involved that you would need ten times or a hundred times more data to make your statistics of any value. I hope that we can learn to benefit, as engineers, by previous experience. All those who have worked on the problem have found that a single number has almost no value in determining what is to be done, either for packaging of equipment, or for the more adequate design of equipment.

Youngs: Damage to shipped items by various modes of transportation may readily be inserted in the Thurstone-type system because you do not need to use the normal numbers of engineering. The purpose of the Thurstone system is to demonstrate which variables are related; once you have obtained that information, you can go back to a more detailed analysis of those variables themselves. I agree with you that much work has to be done, but I really can't say what difference there is between planning to do a lot in one chunk and doing it in little driblets over a long period of time. If you are using peak accelerometers or reed gages, or anything else, you have to have the statistical work performed by somebody.

Markowitz: The point I was trying to make is that the designers of equipment and packaging need certain information to help them improve their procedure and they have found that the single number doesn't help them. If we are going through this all over again, suppose we start with the factors that the engineers need. The single number has not given us those factors.

Youngs: That is the essence of the Thurstone method, to try to find that single factor or that single group of factors. It is very likely to be a multiple thing.

R. E. Blake, NRL: The analysis discussed by Mr. Youngs is evidently aimed at finding which dynamic quantity measured is best correlated with damage occurring during shipment. But there was no mention of recording damage to equipment or of shipping actual equipments. If this were attempted, it would introduce further complications because a "typical" or "average" equipment does not exist. The closest thing available is a reed gage, which is a "representative" equipment. The reed gage contains the essential feature of an equipment, namely vibratory elements, and shows the potential damaging capacity of a shock to each of these elements. The curve of damaging potential vs. element frequency (a shock spectrum) shows how satisfactory it would be to express shock severity as a single number. It is often the case that the impulse (velocity change) of a shock is the criterion of damage for low-frequency systems while acceleration determines failures of high frequency systems. Whether a similar breakdown is permissible in this problem can be seen by inspection of the shock spectra.

* * *

USE OF THE ENERGY METHOD IN THE DESIGN OF PACKAGE CUSHIONS

G. S. Mustin,* BuAer

Considerable attention has been given in recent years to the problems of proper design of package cushions to meet transportation and handling hazards, or, perhaps more correctly, the rough handling criteria established by existing specifications (References 1, 2, 3, 4, 5, 6). Fundamental in the field is, of course, the classical paper by Mindlin (Reference 7).

The Mindlin equations, however, have not been generally adopted by package designers. One possible reason is the inhibitive effect of the differential equations which, at first glance, are too formidable for use in day to day work where a relatively small organization may be called upon to design as many as 200 packages a week.

Accordingly, a simplified method has been sought which would meet the criteria of rapidity and simplicity without too great a loss of accuracy and yet, if desirable or necessary because of failure in test or actual shipment, would permit more complex calculations based on the original data.**

A method which seems, at this time, to meet these criteria is the energy method outlined below.

1. The maximum allowable load on the item is determined by multiplying its weight by the maximum allowable acceleration in g's. This latter factor is usually handed to the package designer. The load is

$$L_{max} = Wa_{max} . \quad (1)$$

2. The energy associated with the drop height is obtained from the equation

$$E = Wh. \quad (2)$$

3. From the load-deflection curve of the cushion material desired for use, the load-energy curve is computed by integration or, more simply, by use of a planimeter or by the expedient of counting squares. These relations are shown in Figure 1 which represents an actual compound mount used by Sprenkle in the design of an aircraft engine container (Reference 8).

4. The load associated with the energy from Equation (2) is read from the load-energy curve, and direct comparison with the value from Equation (1) is immediately possible. This load is divided by the weight to find the maximum acceleration encountered in g's. Comparison can be made on this basis if desired.

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*The opinions expressed herein are those of the author and do not necessarily represent those of the Department of the Navy.

**Some package designers have expressed a desire to be able to handle the cushion design without the necessity for submitting the package to test. From a technical standpoint it is only necessary to point out that even the most precise mathematical formulae available require broad assumptions concerning major parameters (mostly on the safe side) which tend to lead to overdesign. On the other hand, a fairly simple mechanical system to be protected is usually assumed whereas one usually has a very complex system, a factor which can lead to underdesign. The technique discussed here can be considered only as a rapid method for preparing a package which has a better than fair chance of surviving the test.

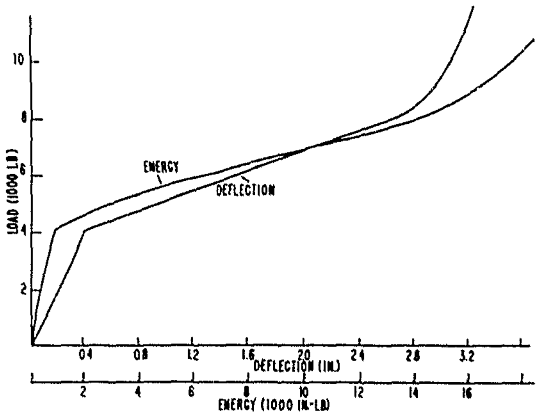

Figure 1 – Load-deflection and load-energy curves for a heavy duty compound isolator consisting of biscuit-type rubber mount and preloaded spring

5. Necessary deflections are read directly from the load-deflection curve and the package design proceeds accordingly.

Slightly different methods of plotting and handling the computations are given by Crede (Reference 9) and Mindlin (Reference 10), but the principle remains the same. In the cited section of Crede's book, there is an example showing the inherent accuracy obtained if damping is not present.

The procedure is essentially the same with diffuse cushions except that loads are expressed in psi and the energy is expressed in in.-lb/sq in. as in Figure 2.

Gretz (Reference 11) has proposed a modification of this technique by substituting the nondimensional term efficiency, as used by the Society of Automotive Engineers (Reference 12). Basically, efficiency is the ratio of the area under the curve to the area if the curve were square as determined at the maximum allowable load point. This method has the advantage of avoiding any semblance of the mathematical terms integration and differentiation; and the disadvantage of being tedious if many computations are required with the same material, since efficiencies will vary from point to point on a curvilinear load-deflection graph. Although the plotting of the load-energy curve is more complex, the curve can be used many times.

Janssen (Reference 13) has refined the energy approach by determining that the minimum point of the curve for force-energy ratio versus force, gives the optimum thickness and load which should be used with the particular cushioning material and has defined the point as J_{opt}. This point is also plotted in Figure 2 which, in turn, is replotted from the curves used in Janssen's paper. Orensteen (Reference 14), has discussed the significance of this point in greater detail. As a minimum application, knowledge of J_{opt} gives a rapid method of selecting the best available material for a given use. For reasons of economy in stocking materials, packages will not always be designed on the basis of J_{opt} and, therefore, the energy approach allows use of any point on the curve. In addition, of course, there are sound reasons for considering a preloaded isolator in the design of which the basic curve is essential. (See Masel (Reference 15) for a discussion of improvement in efficiency by precompression.)

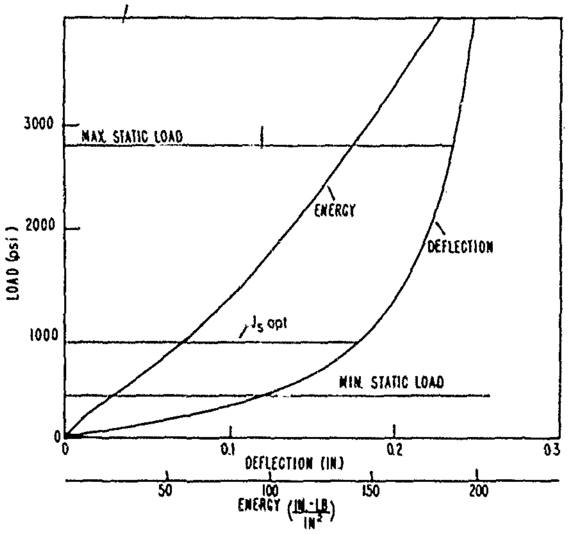

Figure 2 – Load-deflection and load-energy curves for a typical cellulosic cushioning material

The energy method has distinct advantages in the preliminary design of a shock isolator for a shipping container. The method provides a means of quickly designing a package which can be tested with good chance of survival. If necessary or desirable the more complex analysis of frequency effects contained in Mindlin's classical paper (Reference 7) may be undertaken prior to test; or, if failure occurs in the tests, this analysis may be made as a means of determining what should be done to correct the deficiency. All published papers suffer, however, from the disadvantage that few container tests involve only a flatwise drop. Even on small containers, drops are frequently on the corner with subsequent rotation (Figure 3), while with large

containers the drops are definitely rotational. The rotational component may be obtained by container roll-over or push-off from the edge of a high platform, or may be obtained by pivotal drops of the type found in the aircraft engine container specifications (References 3 and 4).

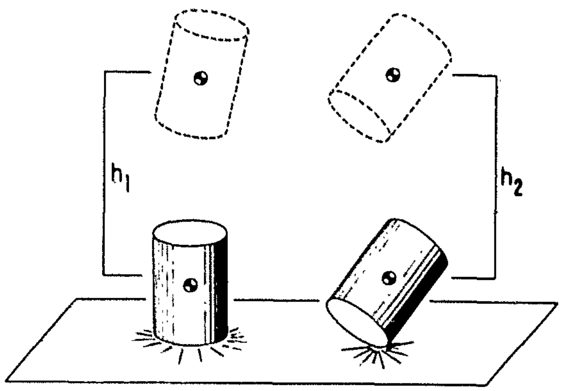

Figure 3 - Typical flatwise and cornerwise drop tests

Flatwise Drop on Any Side or End		Corner Drop on Any Corner or Quadrant	
Height of Drop h_1 (in.)	Container Weight (lb)	Height of Drop h_2 (in.)	Container Weight (lb)
30	250	30	250
24	215-500	24	251-500
18	500		

Experience in the case of aircraft engine container design has shown that the energy method can be used, with resulting designs having a good chance of survival in test and in service. In essence this is the same technique used by Sprenkle which he later subjected to analysis by high-speed movies (Reference 16). Thus the technique is not new although the actual sequence given here may be somewhat different, and to certain persons some of the applications may be new. The procedure given is for a rotational drop of the type contained in Specification MIL-C-5584. Similar geometric analyses to determine height of drop for the other types of tests can easily be performed and subsequent calculations are essentially similar.

The essential geometric relationship of this type of drop test is shown in Figure 4. One end of the container is placed on a block of height h_1. The opposite end is raised to height h_1 and is released to fall freely. The dotted lines in the figure indicate the positions, at the beginning of the test, of the container base and of the center of gravity of the item (point G'). The solid lines indicate positions at the instant of impact. The height of drop, h, is given by

$$h = \frac{h_1 x_1}{L} . \tag{3}$$

Note that x_1 and L are actually in terms of the package. For fairly large items, however, the ratio of the distance between the pivot end of the item and the c.g. to the total length of the item can be used with good accuracy.

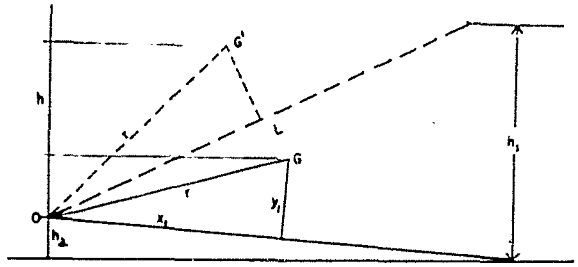

Figure 4 – Rotational drop test--configurations at beginning of test (dotted lines) and at instant of impact (solid lines)

The moment of inertia of the item about the pivot point, 0, is given by

$$I_2 = I_1 + \frac{r^2 W}{g} \tag{4}$$

where

I_1 = moment of inertia about the c.g.
I_2 = moment of inertia about 0
$r^2 = x_1^2 + y_1^2$
W = weight of the item
g = acceleration of gravity

The square of the radius of gyration, k^2, is computed by

$$k^2 = \frac{I_2 g}{W} . \tag{5}$$

With the above values it is possible to determine the total energy in the system, E, and the proportions of this energy which are translational, E_1, and rotational, E_2, by the following relationships:

$$E = Wh = E_1 + E_2 = E_1 \left(1 + \frac{k^2}{r^2}\right) , \tag{6}$$

$$E_1 = \frac{Wh}{1 + \frac{k^2}{r^2}} , \tag{7}$$

$$E_2 = E - E_1 . \tag{8}$$

After impact, the item will be deflected normal to r along the line G-G'' as a result of the translational energy and will tend to rotate about its c.g. from the rotational energy, as shown in Figure 5. It is apparent that the angle formed by the intersection of lines d_1 and d_2 is equal to θ_1, which from Figure 4 is

$$\theta_1 = \tan^{-1} \frac{y_1}{x_1} . \tag{9}$$

Accordingly, the translational energy normal to the base of the box, E_3, will be given by

$$E_3 = E_1 \cos \theta_1 , \tag{10}$$

while the translational energy parallel to the base of the container, E_4, is given by

$$E_4 = E_1 \sin \theta_1 . \tag{11}$$

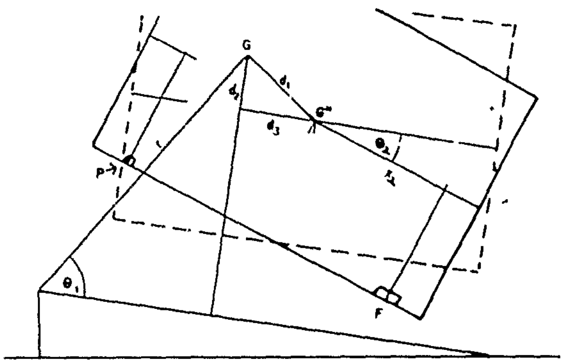

Figure 5 - Rotational drop test--configurations at instant of impact (dotted lines) and at maximum deflection (solid lines)

With the values given by Equations (10) and (11), it is possible to enter load energy curves resulting from load deflection curves of any shape and determine loads, deflections, and accelerations of the c.g. by the simple means previously outlined. For diffuse cushions, Janssen's J_{opt} can be used in cushion selection as heretofore. If the isolator is linear in the operating range, as shown in Figure 1, a situation common with shear sandwiches, these values can be determined by direct computation from the following equations

$$d_1 = \sqrt{\frac{2E_1}{NK}} \tag{12}$$

where N = number of mounts and K = spring rate of one mount;

$$d_2 = d_1 \cos \theta_1 , \tag{13}$$

$$d_3 = d_1 \sin \theta_1 , \tag{14}$$

$$a_{H_G} = \frac{NKd_3}{W} , \tag{15}$$

$$a_{V_G} = \frac{NKd_2}{W} . \tag{16}$$

Before proceeding with construction of the package, however, it is necessary to determine whether the isolator selected will absorb the rotational energy without causing excessive loads on the item or allowing the item to strike the side of the container. To accomplish this, the spring rate is determined by taking the slope of the curve at deflection d_2. A first approximation of the angle of rotation in radians is then given by

$$\theta_2 = \sqrt{\frac{2E_2}{\frac{N}{2} K x_2^2}} . \tag{17}$$

In order to be able to operate directly with Equation (17) it should be assumed that the diffuse cushion is concentrated in a massless column of suitable cross section, usually one square inch, located half way between the center of gravity and the end of the item. Additional parameters should be added, however, if the cushion completely surrounds the item. Assuming that there are two isolators, P and F in Figure 5, the linear deflection due to rotation, R, will be given by

$$d_R = x_2 \theta_2 , \tag{18}$$

with corresponding total linear deflections of P and F of

$$d_F = d_2 + d_R , \tag{19}$$

$$d_P = d_2 - d_R . \tag{20}$$

For nonlinear isolators, it is necessary to take the spring rate associated with d_F and repeat the operations of Equations (17) through (20) until the change in deflection of F in successive computations is insignificant for further design purposes.

With the angle of rotation, θ_2, determined satisfactorily by the above procedures, the angular acceleration in radians/sec^2 of the item about its c.g. is given by

$$\alpha = \frac{2E_2}{I_1\ \theta_2}. \tag{21}$$

Inasmuch as it has been assumed that the item is a rigid body, all distances can be plotted along a straight line as shown in Figure 6. The acceleration normal to the base of the container of any point of distance x towards the impact end from the c.g. will be given by

$$a_{x+} = a_{v_G} + \frac{x\alpha}{g}. \tag{22}$$

while that of any point of distance x to the left of the c.g. is obtained by simple subtraction

$$a_{x-} = a_{v_G} - \frac{x\alpha}{g}. \tag{23}$$

Specifically, the accelerations of the ends of the item and the mount points, as shown in Figure 6, are given by

$$a_A = a_{v_G} - \frac{x_3\alpha}{g}, \tag{24}$$

$$a_P = a_{v_G} - \frac{x_2\alpha}{g}, \tag{25}$$

$$a_F = a_{v_G} + \frac{x_2\alpha}{g}, \tag{26}$$

$$a_B = a_{v_G} + \frac{x_4\alpha}{g}. \tag{27}$$

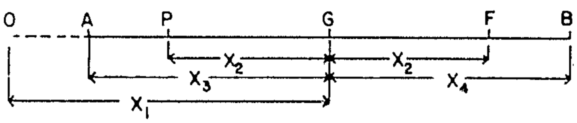

Figure 6 - Distances between various points on item expressed as lengths of projections along the base

Vertical and rotational natural frequencies associated with the deflections encountered in the drop tests are given by

$$f_v = \frac{\omega_1}{2\pi} = 3.13\sqrt{\frac{NK}{W}}, \tag{28}$$

$$f_R = \frac{\omega_2}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{2E_2}{I_1\ \theta_2^{\,2}}}. \tag{29}$$

For linear systems this is a good measure of the natural frequencies under steady-state vibrations. For nonlinear systems, frequencies obtaining with lower energy inputs would have to be computed. This can be done by again using the load-energy curves for the material. If rail transportation is being considered, the data given by Guins (Reference 17) are considered particularly important as a starting point in these computations.

The rotational velocity at impact is given by

$$\omega_3 = \sqrt{\frac{2E}{I_2}} \tag{30}$$

from which the linear velocity of any point at distance x' from the pivot point 0 is given by

$$v_{x'} = x'\,\omega_3. \tag{31}$$

Equations (17), (19), (20), (28), (29), (30) and (31) can be used to estimate positions of the item in the container at any time after impact by some graphical means such as that shown in Figure 7. Such an estimate is useful in determining that rotational and vertical motions are in proper relationship.

The method, in all of its applications, obviously has some serious weaknesses. Among the most salient are:

1. The container and item are assumed to be rigid bodies. The analysis tends to make the actual design achieved quite conservative unless some allowances are made for elasticity of the container.

2. Concentration of the diffuse cushion as a massless spring at an arbitrary distance from the c.g. is obviously introducing unknown errors into the analysis if the material is non-linear, which is true in most applications. Inasmuch as no completely satisfactory method of handling this situation has yet been developed (although it is hoped that current studies at the Naval Research Laboratory will eventually produce such a method) the method given here can only be offered as providing an approach, a first approximation, for design purposes.

3. An inherent error in predicting maximum acceleration on the outer case of the item is lack of a damping term. As pointed out by

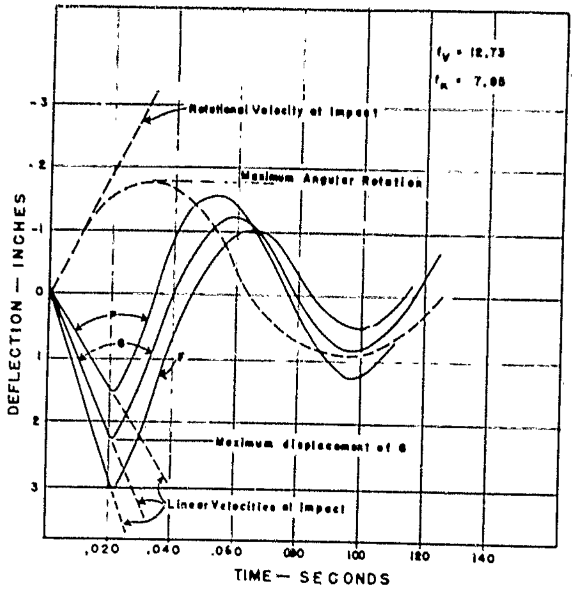

Figure 7 - Plot of Equations (17), (19), (20), (28), (29), (30), and (31) for an aircraft engine container installation. Sinusoidal vibration and approximately 20 percent loss of energy on each cycle assumed.

Mindlin (Reference 18), the method is, therefore, conservative for velocity damping less than 50 percent of critical. For values higher than this, accelerations will be higher than predicted. Another source of error is the lack of a factor expressing the effect of the ratio of the mass of the cushioning to the mass of the item. This has, in general, a tendency to make the predicted accelerations higher than those actually obtained (Reference 19), thus further rendering the design technique conservative. It would, of course, be possible to estimate effects of damping and mass after the preliminary design was completed. Experience and good judgment are required to determine whether such effort is warranted in special cases.

4. The energy method is designed to predict only the maximum acceleration on the outer case of the item. In flat drops, use of the Mindlin approach (Reference 7) to estimate internal response is, of course, always possible. Methods of predicting behavior of elements of the item in rotational drops to the same degree of precision used by Mindlin are currently under study at the Naval Research Laboratory. The method given in this paper would still be valid for most preliminary design purposes, however. In any case, the package designer is rarely given any other value than the maximum number of g's that the item as a whole is capable of withstanding.

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EVENING SESSION

The evening session was held on 12 May in the Smithsonian Institution Auditorium. Dr. W. J. Sette, DTMB, presided.

The work of the session was divided into two parts:

1. An illustrated presentation of "The Shock Tests Against the SS428" was given by H. L. Rich, DTMB. The presentation was made in the following order:

   (a) Description of test arrangements

   (b) Documentary film on test procedures

   (c) High-speed motion pictures of shock motions

   (d) Slides showing shock damage.

2. A discussion on "The Validity of Accelerated Vibration Testing by Increasing Amplitude and Decreasing Time of Test" was conducted by D. E. Marlowe, NOL.

   The discussion featured a review of this important phase of simulation testing and a general discussion of some of the problems involved. However, as accelerated vibration testing appears to be of considerable importance practically, it is desirable to gather more completely the available information relating to it. Both theoretical and experimental work in the direction of accelerated testing have been conducted by various groups but the information needs to be correlated for engineering use.

Therefore, a supplement to this Bulletin will be issued, which will include certain discussions that have been omitted from the evening session. In addition, an attempt will be made in the forthcoming Supplement to present the available information on accelerated testing and how it may be correlated with fatigue failures in structures, with vibration damage which occurs in complex equipments when in operational use, and with data predicting the service-life of structures from fatigue and decrement measurements.

SHOCK TESTS AGAINST THE USS ULUA (SS428)

H. L. Rich, DTMB

These tests were conducted during the summer of 1952. The ULUA was a fleet-type submarine on which construction was stopped at the end of the last war. At that time the hull was essentially complete but only a small amount of equipment necessary for its operation had been installed.

The tests were conducted jointly by the Underwater Explosion Research Division of the Norfolk Naval Shipyard, the Naval Research Laboratory and the David Taylor Model Basin. The main interest of the Underwater Explosion Research Division, which made over-all arrangements for the test, was in the effect of these underwater explosions on the hull itself; that of the Naval Research Laboratory was in the effect of the shock motion on special equipment for nuclear-propelled submarines; and that of the Model Basin was in the study of the shock motion produced throughout the vessel and its effect on equipment in general.

Because of the complexity and cost of running this type of test, this was the first conducted in the United States since the tests on the USS DRAGONET (SS293) in 1944. It is noteworthy that the least severe shock experienced in the test conducted during the past summer was more severe than the most potent shot fired during the earlier DRAGONET experiments. The new tests were highly desirable to obtain data under close attack conditions that might be lethal to a submarine in combat.

Since one of our objectives was to determine the effect of shock on submarine equipment, the existing installations were supplemented by approximately 50 standard items considered representative of submarine equipment in general. However, the total equipment installed represented only a small portion of that in an operating vessel; moreover, the components were not connected into integrated hydraulic, pneumatic, electrical, communication, or propulsion systems. Consequently, the resulting damage was probably neither as severe nor as widespread as would be caused in an operating vessel.

Largely bacause of this lack of operating equipment, the Model Basin test objectives were aimed primarily at obtaining instrumental measurements of the shock motion both on the hull and on weights installed to simulate equipment. Briefly, the DTMB instrumentation included approximately 70 channels of electrical instruments, including velocity meters, accelerometers and pressure gages. The outputs of these meters were recorded remotely in the Model Basin instrument trailer, located on a supporting vessel, the UEB-1. In addition, various types of autographic instruments were installed. These included approximately 60 multifrequency reed gages, 300 peak reading displacement gages, and two high-speed motion picture cameras. The reed gages were used to obtain shock spectra. In the DTMB MK IV Multifrequency Reed Gage (Figure 1), the maximum excursions of the five weighted reeds on each side of the gage are recorded on waxed paper. The response of the reeds is characteristic of the response of simple mechanical systems having the same natural frequencies. A time history of the reed motion may be obtained by adding a motor and special paper drive assembly. The displacement gages were used in groups to record the peak displacements of equipment on resilient mountings.

With the exception of four preliminary tests in the Norfolk Naval Shipyard, the trials were conducted in the Chesapeake Bay in the vicinity of the Naval Air Station, Patuxent. All charges were detonated with the submarine submerged to periscope depth in 105 or 145 ft of water. Fifteen charges were detonated at distances varying from 75 to 30 ft from the pressure hull. Each charge contained 250 lb of HBX, which is equivalent to 400 lb of TNT. Shots were fired off the control room, off the crew's quarters compartment, and off the forward engine room and at various orientations such as might occur in combat.

The material presented below consists of a series of figures which show the response of equipments in the vessel to shock as recorded by high-speed movie cameras, and a selected group of figures which show equipment damaged during the course of the tests.

Figure 2 is an aerial view of the test array. The photograph shows the SS428, the UEB-1, and the YC-1060. Note the instrument cables from the SS428 to the UEB-1. These lead to the instrument trailers on the top deck as well as to an instrument house on the lower deck. The YC-1060 in the left foreground was used to supply compressed air and to control the raising and lowering of the submarine; the air lines can be seen in the figure.

A novel technique was used to record the response and damage to equipments in the vessel. The setup consisted essentially of a "sky-hook" suspension for the Eastman high-speed camera (Figure 3), mounted in the crew's quarters compartment. The camera, lights, batteries, and control box (behind the batteries) were mounted as a single unit and suspended by a rubber shock cord from the overhead in the submarine. With this mounting the unit has a vertical natural frequency of approximately one cycle per second and the camera was essentially isolated from

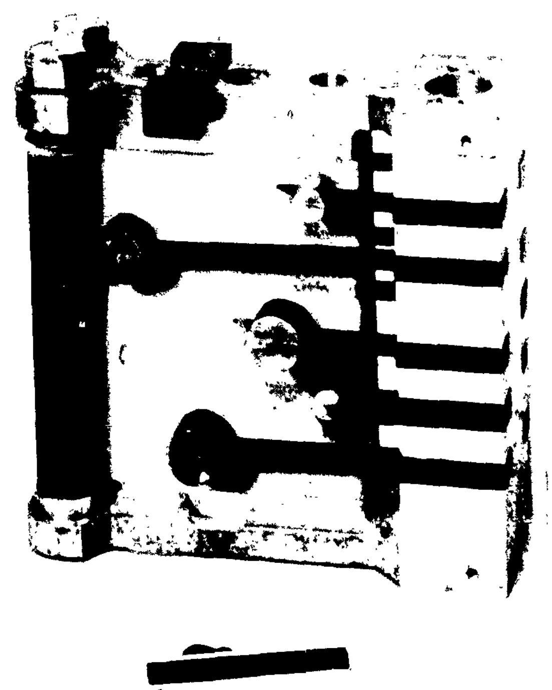


Figure 1 - DTMB Mk IV multifrequency reed gage

Figure 2 - Aerial view of test array

the shock motion of the hull. Consequently, the pictures show the phenomena which occurred immediately after the arrival of the shock wave and are not blurred as they would be if the camera was more rigidly attached to the hull. The positioning cords on the bottom are made of softer rubber than the shock cord.

Figures 4 and 5 contain selected frames from high-speed motion pictures showing motion under shock of a Loran console attached by commercial rubber mountings to a shelf welded to a periscope well (at the right) in the control room. In Figure 4, the shot which was fired alongside the submarine 37.5 ft from the pressure hull at the left produced an initial displacement of the hull to the right. Consequently, the initial pictures after the shot show a motion of the equipment to the left relative to the shelf, followed by a rotation of the equipment in a counterclockwise direction. The plug-in interconnection cable partly seen below the shelf has come out and is falling free. In Figure 5, the latches holding the right-hand chassis to the sub-base were not secured during the test. The first separation of the chassis from its bases (Frames a - h) was produced by the shock wave. The second (Frames i - p) was due to the gas globe pulse.

Figure 6 shows a 4000-lb weight resiliently mounted from a foundation attached to the pressure hull in the crew's quarters compartment. Note the initial violent motion, due to the shock wave, of the hull relative to the weight, and the response of the weighted reeds of the reed gage attached to the weight. This motion was followed by a more gradual translation, due to the expansion of the gas globe produced by the explosion. We believe that it was the early rapid motion of the hull, characterized by a violent rise in velocity at a rate of several thousand g's, which was generally responsible for equipment damage.

Equipments on which shock damage was sustained are shown in Figures 7 through 17. Figure 7 shows a main ballast tank valve used in flooding and blowing ballast tanks. During the tests the main casting on this and other similar valves failed, making the flapper, which is pivoted on the main casting, difficult to operate. This damage would make it difficult to bring the submarine to the surface. In addition to this type of failure the contact maker attached to one of these valves

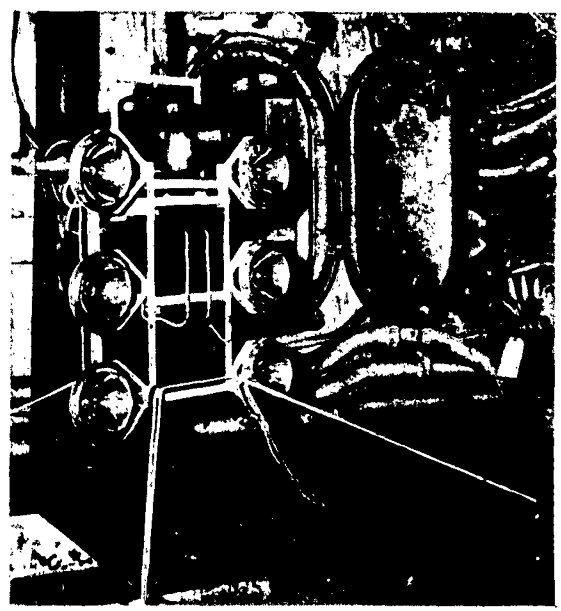


Figure 3 - "Sky-hook" mounting for high-speed camera

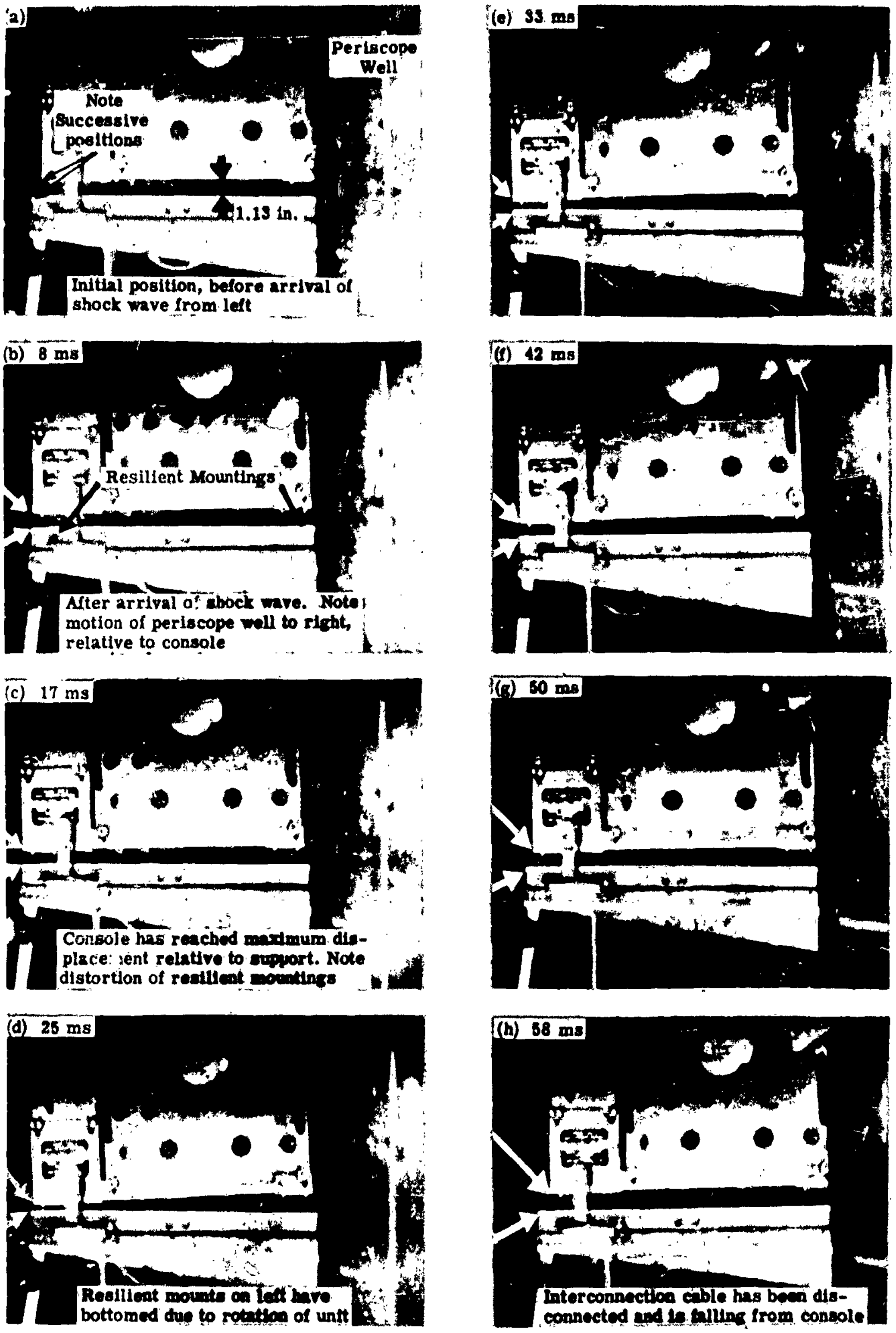


Figure 4 - Initial motion, during Shot 11, of Loran console

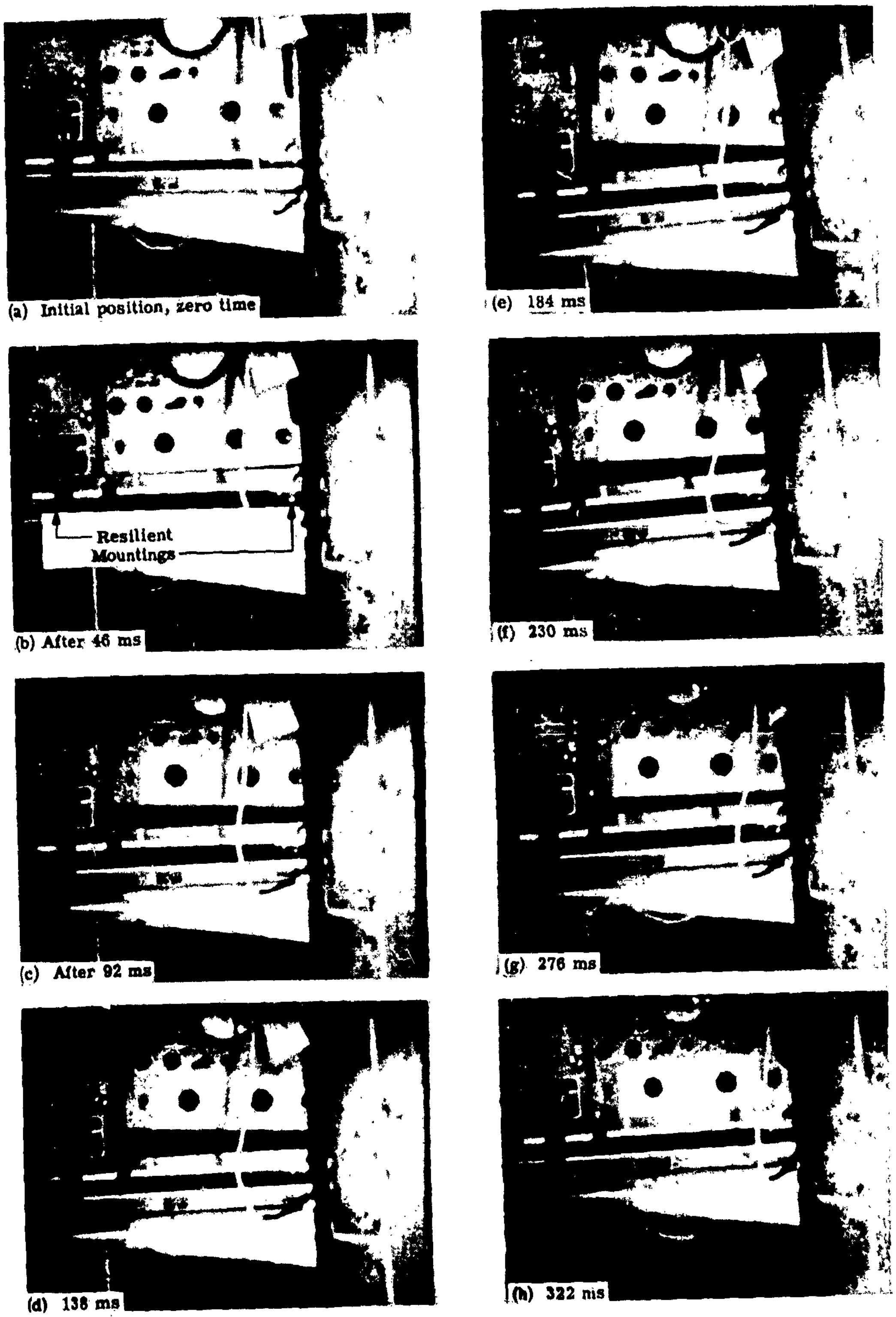


Figure 5 (a-h) - Sequence 1: Initial motion, during Shot 2, of Loran console

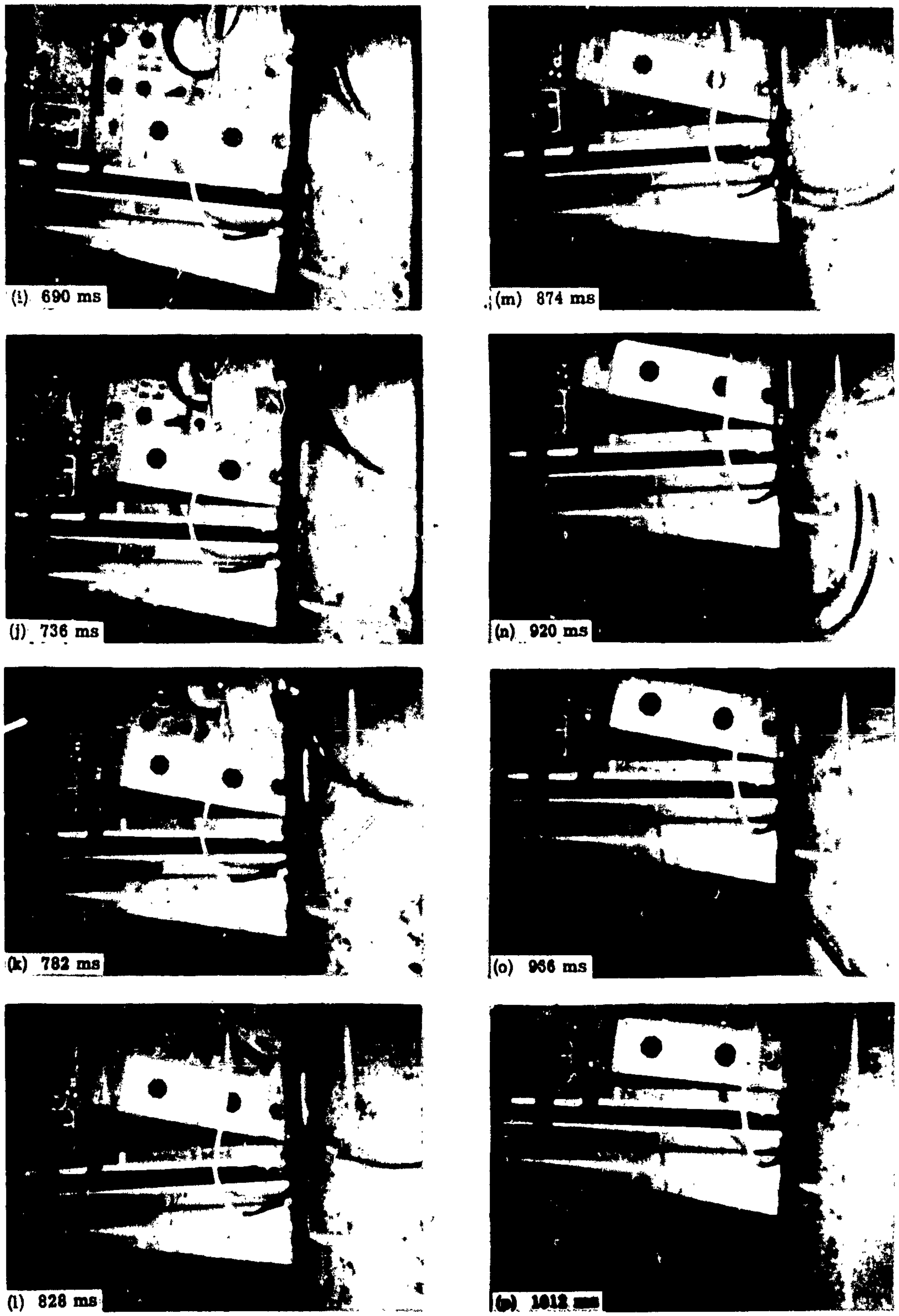


Figure 5 (i - p) - Sequence 2: Motion of Loran console due to shock wave from globe after Shot 2

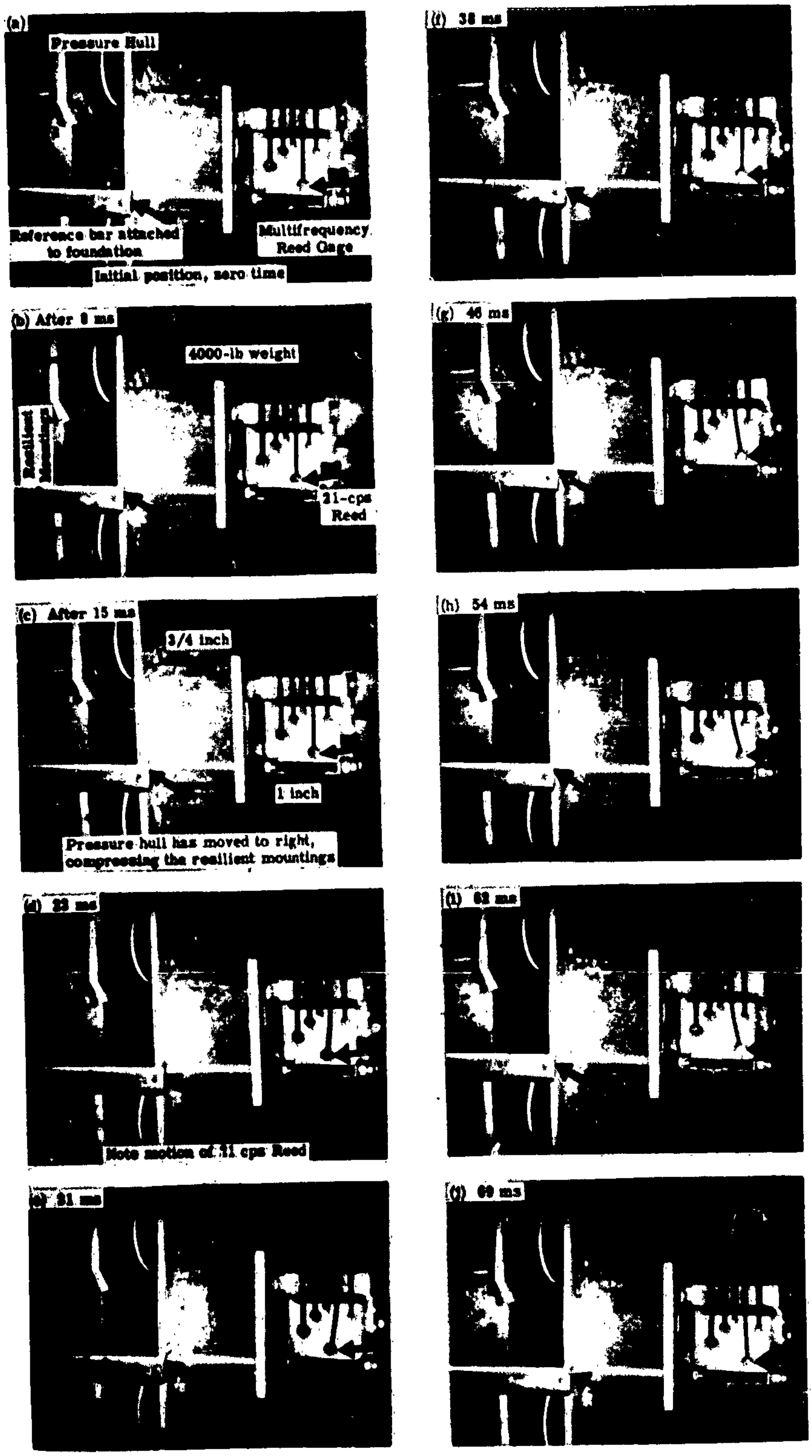


Figure 6 - Motion of 400-lb weight after detonation of charge to left abeam of submarine

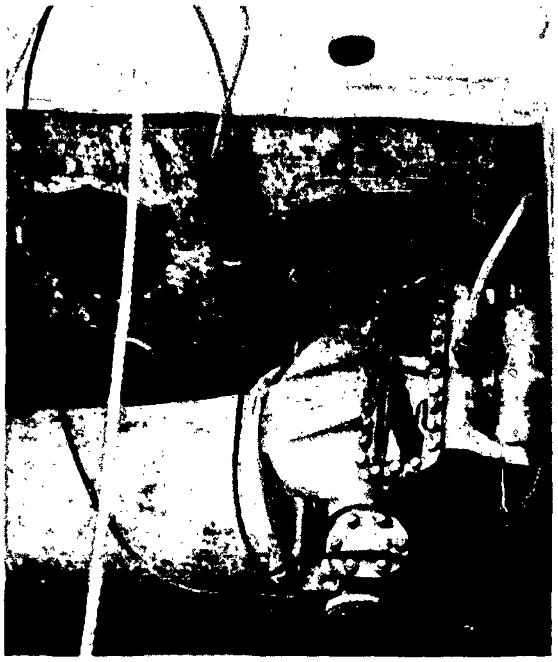


Figure 7 - Main vent valve for No. 5 saddle tank, starboard after Shot 6

broke loose due to failure of hold-down bolts. Inasmuch as the contact maker is a switching arrangement which indicates on the Christmas tree in the control room the condition of the valve, the reading on the Christmas tree could therefore be incorrect.

Figure 8 shows the flapper valve assemblies in one of the ship's bulkheads. The valves are a part of the ship's ventilation system. Watertight integrity is maintained when the ship is in battle condition by closing flapper valves on either side of the bulkhead. The two flapper valves on the left have broken loose due to failure of the brass pin which attaches the flapper to the operating mechanism. A modified type of valve is being tested in the forthcoming SS428 tests this summer.

Figure 8 - Ventilation valves in bulkhead after Shot 12

Figure 9 is a photo taken after Shot 12 of the ship's depth gage, a 10-in. Bourdon gage. The gage was fastened prior to the test to a steel panel by means of bolts and felt washers. As a result of the explosion the top bolt failed, and the brass case was torn at the other two bolts. The gage was inoperative due to internal derangements. In other Bourdon gages there were failures of plastic cases, cover glasses, and mechanisms.

In the center of Figure 10, two plastic intercommunication connection boxes are shown. These boxes have inserted ears which were bolted to the hull. During the tests the boxes were projected from their attachment ears by shock motion of the hull. On the right-hand side of the picture is a telephone call and ringer box, one of which is located at each compartment of the vessel. The box contains a magneto generator which was made inoperative by loss of a gear that turns the magneto. We found that during the tests the magnetos failed progressively at all locations except in the forward engine room, Figure 11. At all except this location the box was bolted to the hull. In the latter case, however, the two connection boxes and the ringer box were attached to a panel which was resiliently mounted on felt mounts from the pressure hull. In this case the attentuation of the shock motion was sufficient to prevent damage.

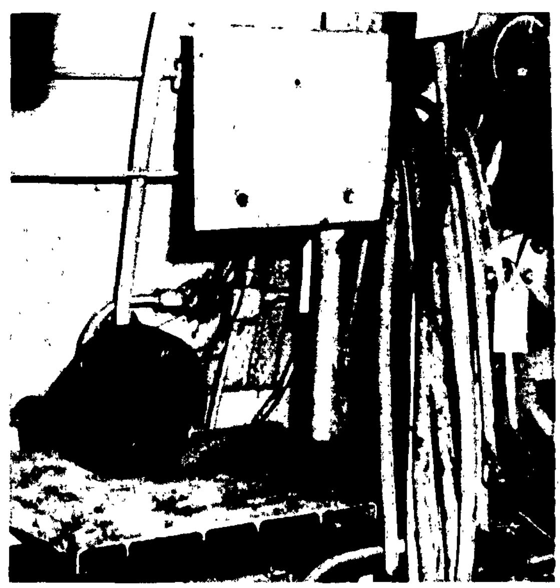


Figure 9 - Fathometer in central room after Shot 12

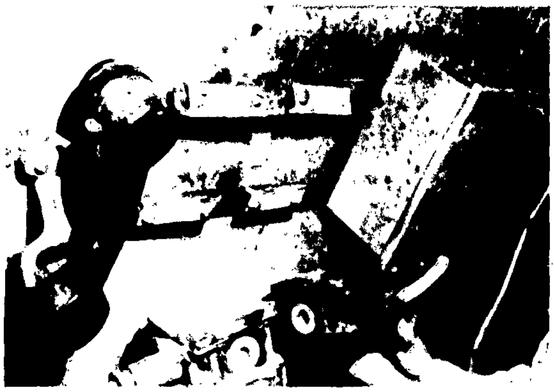


Figure 10 - Broken intercommunication plug boxes and telephone ringer in officers' quarters compartment after Shot 12

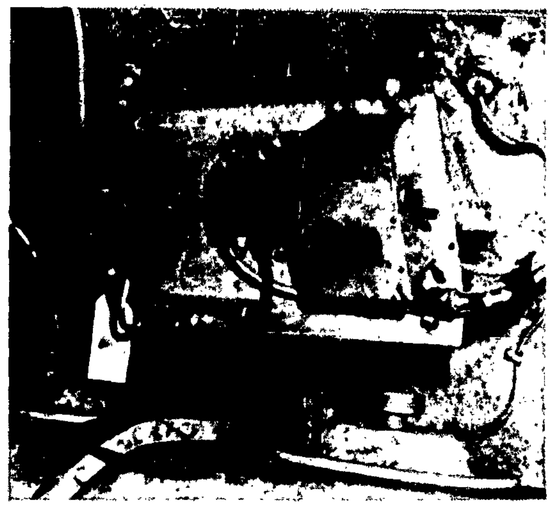


Figure 11 - Undamaged intercommunication plug boxes and telephone ringer in forward engine room

A modified type of ringer will be tested in the next series of SS428 tests. In addition to failure of this type of connection box, there were also failures in a cast aluminum type of electrical connection box and in a steel box having spot welded hold-down strips. Failure in the aluminum box was by fracture of the aluminum casting; failure in the steel box was by failure in the welds.

Figure 12 is a photo taken, after Shot 12, of one of the two main engine lubricating oil coolers which were on the ship during the test. The cooler which normally contains fluid was empty during the tests; it had been attached to the bulkhead at the forward end of the after engine room by means of twelve, 5/8-in. bolts. There are normally three bolts on each side near the bottom and three more on each side at the top. In the figure, which does not include the top of the equipment, you see the bottom bolt flange on the inboard side of the cooler.

After Shot 12 it was found that the cooler was supported by only the three bolts at the lower outboard side; the other nine bolts had failed. We wondered whether this failure might not have been partially due to progressive stretching of the bolts during the shots which preceded the one in which this failure was noted. Consequently, we substituted new bolts for those that had failed. We found in the following test that the failure was repeated. During forthcoming tests a different bolting arrangement will be tried.

Figure 13 shows the pump mounting plate, part of the port side main diesel engine. In this case, all 32 bolts which held the pump plate to the diesel engine failed. Consequently, the pump plate had dropped and the engine was inoperative.

Figure 14, a closeup view of a part of the pump plate, shows one of the stub shafts of the engine. The misalignment between the shaft and the pump plate is evidenced by the failure of the shaft core.

Figure 12 - Port main engine lubricating oil cooler, showing bolt failures

Figure 13 - Gear case and pump mounting plate of port main diesel engine showing bolt failures after Shot 17.

Figure 15 is a photo of one of the two main generators with which the SS428 was equipped during this test. The main generators are driven by the diesel engines. After Shot 16 we found that the four 1-1/8-in. bolts which held the generator to the foundation had all failed.

Figure 14 - End bearing of stub shaft for driving fuel oil pump of port main engine after Shot 17

Figure 15 - Main starboard generator in after engine room showing bolt failures

Figure 16 - Damaged cathode-ray tube from Type AN/BQN fathometer in control room after Shot 14

Figure 16 shows a cathode-ray tube which was in the ship's electronic fathometer. The tube base has come loose from the tube itself and there is a considerable amount of distortion of the shield at the left side of the photograph. In general the amount of damage to electronic equipment, including those items which were not rigidly mounted, was surprisingly small.

Figure 17 shows one of the lights of the ship's emergency lighting system. The damage is rather obvious. A number of types of fluorescent fixtures were also tested. In some fixtures the socket which held the fluorescent lamp broke permitting the lamp to be projected out of the fixture.

In conclusion there are several points worthy of emphasis:

1. Although the ULUA contained only a small fraction of the items of equipment necessary to the operation of a submarine, a sufficient number of failures occurred in the equipment

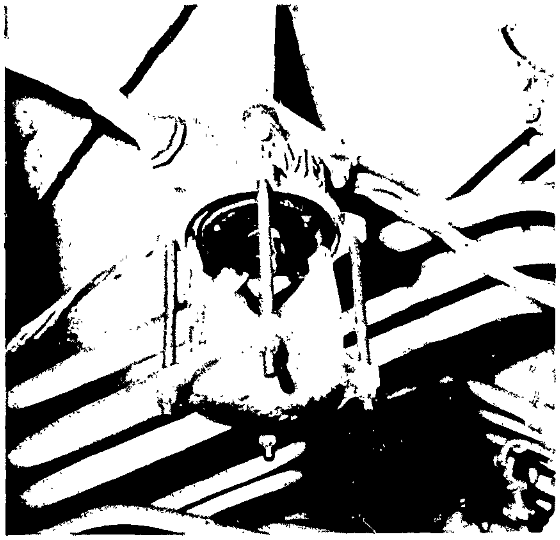


Figure 17 - Emergency light after Shot 13

on board to render the submarine inoperable. For example, failure of a vent valve would have made surfacing difficult; failure of the main engines would have immobilized the vessel at least temporarily, and failure of the bulkhead flapper valves would have destroyed watertight integrity. It is reasonable to conclude that if more equipment had been installed damage would have been more widespread.

2. During this entire series of tests no flooding of any compartment occurred. This indicates that the equipment on a submarine is more vulnerable than is the hull. Consequently, a considerable effort must be made to bring the strength of equipment up to that of the hull.

3. This test points up a need for full-scale tests on a modern operating vessel. Such a test, properly conducted, will permit a more accurate evaluation of the shock problem on submarines and will in addition show up specific weaknesses in equipment.

This discussion represents only a small portion of the information obtained by the Model Basin during these tests. Some other results were presented in an earlier paper by Mr. Ruggles. At the present time the instrumental data are being analyzed and correlated with equipment damage. This information will be forthcoming in DTMB reports.

DISCUSSION

H. M. Forkois, NRL: I was wondering if any attempt was made to correlate the damages occurring in the submarine test with actual simulated tests in the laboratories, such as are conducted at NRL.

Rich: The correlation of damage produced in the laboratory on shock machines with damage occurring in submarines is of great importance and is being studied wherever possible. Unfortunately, the number of items on the submarine was very limited and only a small fraction of these had been previously tested on Navy shock machines. Of the few that were tested on the machine, some survived intact while others were damaged in the ULUA tests.

Dr. M. G. Scherberg, WADC: This afternoon I think we heard a paper to the effect that stiff mountings are desirable on shipboard. It seems to me that you have material here for a thesis to the contrary.

Rich: I believe that this paradoxical situation exists because there is no single optimum mounting method to cover all equipment for all locations in all types of ships. Whereas a rigid mounting may be desirable in one situation, a very soft mounting may be called for in another. For a particular piece of equipment the mounting in the superstructure of a vessel may call for a rigid fastening to the ship, while mounting on the hull below the waterline may necessitate a soft mounting. This morning Mr. Ruggles presented a shock spectrum computed from a typical velocity-time oscillogram recorded on the pressure hull of the ULUA. This spectrum shows that while very high accelerations were produced in very stiff mechanical systems, the accelerations in systems having natural frequencies of less than 200 cps were less than those recorded on the hull, the maximum acceleration decreasing rapidly with decrease in frequency. Consequently you would expect that equipment on resilient mountings would be more shockproof, and from the small number of examples on the ULUA, it was.

Scherberg: Was this given in the thesis?

Rich: No, in a paper by Mr. Ruggles.

Scherberg: Were they linear mountings?

Rich: No. Almost all the mountings were either nonlinear commercial or Navy rubber or felt mountings. However, commercial shear type mountings were used in one or two installations.

Forkois: With regard to the thesis referred to by Dr. Scherberg, we have electronic equipments which pass the 40T9 (that is the Bureau of Ships Specification for Shock and Vibration) without shock mounts. The greater proportion of these equipments are installed in surface vessels and in locations where the shock motion is considerably attenuated. My own philosophy in the matter is that if you can do it without shock mounts by all means do so, because vibration in certain parts of a ship can be very severe and more detrimental to equipment than shock. This is due to amplification, particularly of the "rocking modes" when rubber mounts are used.

Now, with regard to the measured frequencies which indicated that below 200 cps a shock mounting system would be desirable, I would say that with rubber mounts supporting a piece of equipment, it is a very difficult problem to get the natural frequencies of the rocking modes out of the shipboard frequency range, that is, above 23 cps. We would like to get these systems higher than 30 cps if possible. To get values in the order of 80 cps would require what is commonly called "solid" mounting. To achieve natural frequencies of about 80 cps with rubber mounts requires such a high degree of stiffness that the rubber becomes so hard that eventually you do not have what is ordinarily considered a resilient mounting. Actually, the information presented seems to confirm the contention that adequate shock protection may be achieved by relying on the "structural" resilience of the equipment mounting system.

Rich: I would like to repeat that the curve presented by Mr. Ruggles showed that for an equipment with a natural frequency less than 200 cps the acceleration was attenuated; i.e., the acceleration of the equipment was less than the acceleration of its foundation.

Scherberg: I am not taking any sides. I am merely pointing out there seems to be some confusion about the information and even in surface ships you have to expect some shock damage in battle.

Forkois: Generally speaking, the Shock and Vibration Program is trying to prevent avoidable damage. Naturally, if a sixteen-inch shell hits a deckhouse and obliterates everything, we can't protect for such occurreaces.

With regard to these 50-fps velocity changes that were measured, I might say that if this is the true condition then the experiments we perform in the laboratory are not similar. The velocity changes that we get on the medium-weight machine for equipments weighing over 200 lb are from eight to ten fps in approximately one millisecond for the initial pulse, and about twice these values for the reverse pulse. Other values are obviously extreme, and I don't know whether we can actually design for 2000 g's.

Rich: If the designer takes advantage of the very short duration of the impact by making his equipment resilient, the accelerations will be attenuated and the equipment will not have to withstand the high peaks.

E. C. Taylor, Portsmouth Naval Shipyard: To get back to the question of damage of equipment on the submarine, I think the comment was made that we should improve the equipment inside the boat for that it will remain operable. I think it should be pointed out that the submarine was very severely damaged. These tests were made at periscope depth and I believe it is true that if we had submerged like a submarine the ship would never have surfaced, because once you lose the circularity of the hull you no longer have a submarine that can submerged to any considerable depth. If we design to the extreme shocks we would have a lot of over-designed equipments in the ship and we don't need that.

Rich: I think we have a long way to go before the equipment in the submarine is equal in strength to that of the hull. In the last slide I showed a picture of the extreme deformation of the hull. The deformation resulted from a premature failure of welds fastening the hull frames to the plating. It may have been due to defective welding. The fastening is being studied and new methods are being tested in the forthcoming ULUA tests.

In any case, even this large deformation did not produce a hull failure. It is generally accepted that the distance a charge has to be from the hull to produce severe equipment damage does not vary with depth. On the other hand, the view is generally held that the lethal radius for the hull increases rapidly with depth; that is, a particular charge is far more effective when a submarine is at operating depth than when the submarine is near the surface. This effect is due to the additional hydrostatic loading of the

hull. Therefore, the argument is made that while the equipment is weaker than the hull near the surface, at operating depth the hull is weaker than the equipment.

There is little data available in this country on the variation of lethal radius of a charge with depth of submergence of the vessel. However, British test results indicate that the increase in lethal radius with depth is small. Consequently the difference in the lethal hull radius near the surface and at operating depth is not enough to change the relationship of hull strength to equipment strength.

Taylor: I am not familiar with actual experiments to obtain data on the change in lethal radius. However, I would like to refer to the experience that we at Portsmouth have had. We are making hydrostatic tests of hulls that have already been damaged by explosions and have found that the strength of the hull once it has lost its circularity is reduced somewhere near 50 percent. The hull with a nine-inch deflection would be in that class, all right.

Now, as to whether the hull was a suitable hull or not, we have run other model tests where the hull pulls away from the frames in the same way. I admit the welding is not perfect, but you can never get a perfect welding job. We can't at Portsmouth, and I don't think we are any worse than the others I know of. It is very difficult to assure that all the frame welds are perfect.

Rich: Because of the safety margin between collapse and operating depth a reduction in hull strength of 50 percent would not necessarily produce hull collapse at the latter depth.

CDR D. Saveker, Armed Forces Special Weapons Project: My opinion is that the test geometries employed when this damage occurred were clearly outside the ranges in which we had expected any sort of hull frame to pull away. I believe that we had a particularly weak hull in the area that was under attack; however, this condition should not be regarded as typical of submarine construction. I think we can do much better, somewhere in the order of 30 to 40 percent better in terms of stand-off.

Unfortunately, you never get a new target to test, at least we haven't so far; the submarine we were testing was a hulk, and, perhaps the mass loadings in the frame were not typical of those in new vessels. However, preparations are in hand for conditioning of tests where we expect to at least have ranges in the trials that produce damage like the type you have seen here.

Perhaps the next time we have a meeting, if this problem comes up we will have some facts from which we can draw conclusions regarding hull damage, in a little better fashion than we have been able to do tonight.

Forkois: I think, perhaps, we ought to define what shock mount and vibration isolators are. A shock mount is a device for absorbing energy at a high rate and dissipating it at a lower rate. A vibration isolator is a "soft" mounting and is used to isolate a steady-state vibration. Generally, the two are incompatible. Recently a noise factor has been added, that is, a noise isolating factor; and it has been found that the very low frequency mount attenuates noise. However, the point which was made, and I think it was brought out in the paper this morning, is that the shock mounts have a tendency to produce a false sense of security; that is, the idea of a black box approach (where the contents are ignored) and applying shock mounts towards the end of a development program to the equipment and expecting the equipment to go through tests just because shock mounts have been placed on it is erroneous. Obviously, if you want the shock mounts to deflect under a shock load, the structure will have to be much stiffer than the shock mounts, or the structure will deform and not the shock mounts.

You must design for stiffness rather than for strength. This may seem a little strange, but it is true. If you consider a square section cantilever beam under static loading conditions, you will find that by doubling the depth of the beam you increase the stiffness eight times, making the natural ffrequency go up by the square root of eight and you would also find that the stress at the section of maximum bending moment will be reduced four times. With respect to the response of the cantilever to a particular shock motion, the dynamic load factor decreases and approaches "one" as the natural frequency is raised. To increase stiffness is not too extremely difficult to accomplish. It is surprising what a few pounds of additional material in the right places will do for an equipment. If you design for stiffness (using materials of proper ductility) and high natural frequencies, the fear of overdesign as related to static loading stress values will be minimized.

The tests we perform are high impact tests which are conducted by dropping a hammer

against a mounting system to which the equipment is secured. A drop test, such as required for portable equipment, where you drop equipment three, four, or five feet on a hard surface and expect the equipment to survive, is a rather difficult thing to design for. Drop tests of that nature are very damaging, particularly if the area of contact is localized.

VALIDITY OF ACCELERATED VIBRATION TESTING BY INCREASING AMPLITUDE AND DECREASING TIME OF TEST

D. E. Marlowe, NOL

I believe we will all agree that accelerated vibration tests are desirable because of the saving of time and manpower. If we want to save time in testing, we may independently vary three main parameters: amplitude, frequency, and duration of test. Extreme variation of temperature may be useful in some cases.

We must first determine the general range of values of these parameters from service conditions, being careful not to overemphasize individual detailed vibration spectra. For conventional vehicles, this work has been done fairly well; more measurements are needed on new forms of transportation (e.g., jet aircraft, guided missiles).

Then we must develop some form of laboratory vibration test that will show whether an equipment is likely to stand up under field conditions. At one time, I became interested in the idea of developing equipment to record acceleration-time histories in the field, and play them back directly on a piece of equipment under test. Considering the wide range of variables we have to deal with, I believe now that this was the wrong approach, and that general purpose equipment should be developed to cover the general range of the variables.

It is really surprising how little direct correlation there is between damage occurring in laboratory vibration tests and damage in the field. In general, the actual power available in the mode of transportation itself, is always far larger than that in the simulation equipment. A little speculation on the problem has made me realize that, after all, we are not in the business of developing vibration tests. We are in the business of building equipment to resist vibration. To whatever extent we succeed, we will reduce the number of cases in which correlation can be obtained. Except in the case of fatigue-type failures, I have not been able to find much data in records from laboratories other than my own to support this view.

However, the decreasing frequency of certain types of failures over the years indicates to me that the form of specification and test which is in common use at the present time, is resulting in a good general screening out of faulty devices.

At this point, I would like to separate vibration failures into two main classes. In the first, structural members are subjected to extended periods of vibration fatigue and the structure itself fails. The second and more common type I have called a "machinery-type" failure for want of a better name. Here, failure is not due to overstressing, but to fastenings coming loose, to internal impact between parts, to dust getting into the parts, or to rubbing of the gears where there is a little play in the gear train.

At present, most attempts to explain the meaning of accelerated fatigue tests include some assumptions as to the degree to which the critical structural member is stressed at each of the frequencies which compose the vibration schedule. This is usually done rather arbitrarily, and while the results are often illuminating, I am not yet convinced that there exists a really sound basis for extrapolation of the results of accelerated tests. I have some hope for an eventual understanding of the meaning of accelerated tests in fatigue-type failures, but I believe that machinery-type failures are too complex, and that accelerated testing can be only a screening operation for the latter.

In our equipment development business, almost all of our failures are of the machinery type. Therefore, our attention must be devoted to using vibration tests to screen out most of the failures in the field. In one respect, this turns out to be a perfectly satisfactory practice. I feel that if you take complex pieces of machinery and subject them to a "stiff" vibration test—one which ranges along the upper boundaries of the amplitude-time-frequency curves such as are measured in the field—you can without too much difficulty and too much overdesign, construct complex pieces of machinery which will survive rigorous field conditions. I hesitate to say that very strenuous effort should be made to pin down such a complex subject more accurately than we understand it at present.

DISCUSSION*

Ben Reznek, NBS, Corona: Since a paper will be given tomorrow morning which takes the opposite stand of what has been expressed tonight, I won't go into any detail tonight except to ask one question. I would like to ask Mr. Marlowe whether he believes that we know enough about transportation and handling shock as well as vibration.

Marlowe: I think that there is still far more to be gained by exploration in the field of shock than in my field of operation.

I would like to point out here that my discussion should not by any means indicate that I discourage the understanding of these things from a scientific point of view. I believe, however, that the great bulk of our problems must be solved by engineering methods.

F. F. Vane, DTMB: I agree substantially with what Marlowe said as to the categories of vibration testing.

I would like to make a few comments, however.

First, shouldn't all our vibration specifications be re-examined so that we can reduce the time of testing to a minimum? For example, do we want to find out what the response of a particular structure is so that the design can be improved?

Secondly, how long do we want to vibrate a structure to see whether these so-called machinery failures will occur?

Thirdly, I believe we should try to keep the vibration aspects separate from the fatigue aspects. In a fatigue test, there are many considerations to be taken into account in order to know just exactly what is happening, particularly in complicated components such as those for use in ships.

I believe that we should become more familiar with methods of operational research, and of quality and quantity control. We need to use additional techniques to study the data already accumulated so that it may be used for better design of equipments.

*Certain discussions have been omitted. These will be included in a supplement to this bulletin, which will be issued soon.

* * *

Closing Remarks

Dr. R. D. Bennett, Dept. of the Navy

I think it is worth saying that this meeting represents a most effective means for rapid, easy exchange of useful, current information among the Services and the various people working on our defense program. It has been a pleasure to have been here and to have had a very small part in your symposium; you are to be complimented on the efficiency with which this organization operates.

I believe it is in order to convey our appreciation to our hosts the Naval Research Laboratory, and particularly to Dr. Klein, who has been so important in the organization of the symposium.

* * *

EXHIBITS

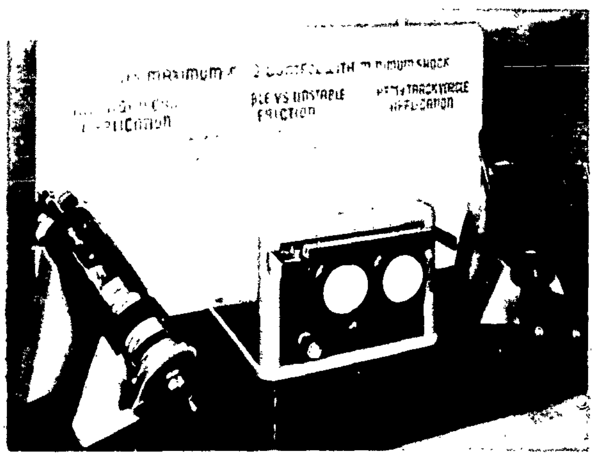
Several carefully arranged exhibits were on display when more than 400 invited scientists and technicians attended the 20th Shock and Vibration Symposium. The exhibits had a common purpose in showing new designs and in some cases new applications for instruments and equipments used in shock and vibration work. The various equipments on display covered several fields of interest such as the measurement of shock and vibration, the control of shock and vibration, pick-up calibration, and simulated tests.

Among the more recently developed equipments was a design for constant friction snubbers. The associated equipment in this exhibit demonstrated the effects of a constant coefficient of friction with changing velocity. Also on exhibit as a new development was instrumentation used for the investigation of strains in ships at sea. The interest in this particular exhibit was heightened by a large display board with the title "Storms at Sea" and a well-arranged group of pictures and pamphlets on the instrumentation and its use.

Other exhibits featured a portable 100-ft drop tester with visual presentation of the waveform of the shock; a variable-cross-section vibrating bar, designed to eliminate the principal causes of failure in vibrating bars; telemetering equipment which brought out recent progress at NRL in telemetering technique; and a new high-speed movie camera and lighting equipment, which demonstrated the use of high-speed photography in shock and vibration studies.

CHRYSLER EXHIBITS

An exhibit presided over by R. N. Janeway of the Engineering Division, Chrysler Corporation included an apparatus designed to demonstrate the nature of the variation in friction coefficient with velocity between any two materials.

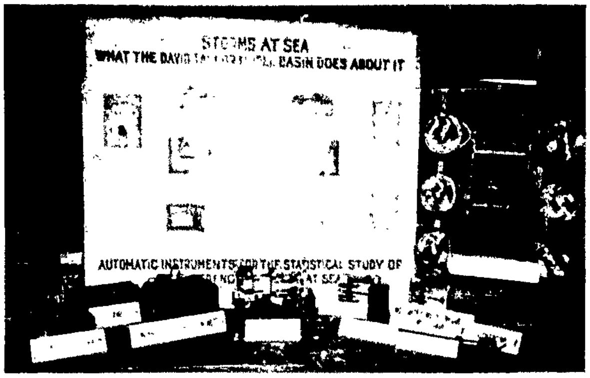


Demonstration apparatus and constant friction snubbers

The apparatus consists of two drums geared together and driven counter to each other at the same speed, with the left-hand drum rotating in the clockwise direction. A bar placed across the two drums and given an initial velocity will obviously be forced to move at different relative velocities with respect to each of the two drums. If the coefficient of friction is independent of velocity the action of the bar will be stable. Once disturbed from its center position it will oscillate at the same or gradually diminishing amplitude. However, if the coefficient decreases with increasing velocity, the action of the bar will be unstable and a self-excited vibration will be induced until the bar leaves the rolls or is otherwise restrained. It will be noted that the drums have two adjacent grooves so that the action of any two materials can be compared simultaneously. When a bar lined with special composition material is placed on the drum side by side with an unlined steel bar the stability of the lined bar and the instability of the unlined steel bar are immediately apparent. In fact, the unlined steel bar is self-exciting without external disturbance.

The left-hand unit in the photograph is a full-size Chrysler design constant friction snubber as applied to railroad cars, with the barrel cut away to show the inner construction. This unit employs the special composition material referred to above. The right-hand unit is a similar snubber adapted for tank installation. The internal construction is practically identical in both units and only the length of travel and external connections are different to suit the particular installation conditions. This unit has now been specified by Army Ordnance for application to the M-48 tank in place of the hydraulic shock absorbers previously used.

DAVID TAYLOR MODEL BASIN EXHIBIT



Display board and equipment

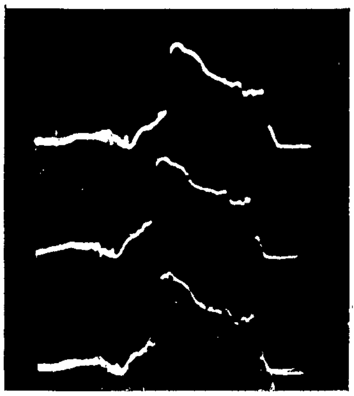
A display board and equipment from the Model Basin were put on exhibit at NRL by C. H. Kinsey, Structural Mechanics Laboratory, DTMB. The operation and application of a group of shock and vibration measuring instruments were shown. Arranged on a table in front of the display board were a TMB Type V pallograph, a General Radio vibration meter, a Brush amplifier and a Brush amplifier and a Brush inking oscillograph. Two types of TMB multifrequency reed gages, a TMB velocity meter, a bolt dynamometer, a shock-mounted high-speed movie camera, and two barium titanate accelerometers were also on display. One of the interesting features was the arrangement on the display board of pictures and pamphlets bringing out the use of these instruments by the Model Basin as it carries forward studies on the strains in ships at sea.

NAVAL ORDNANCE LABORATORY EXHIBIT

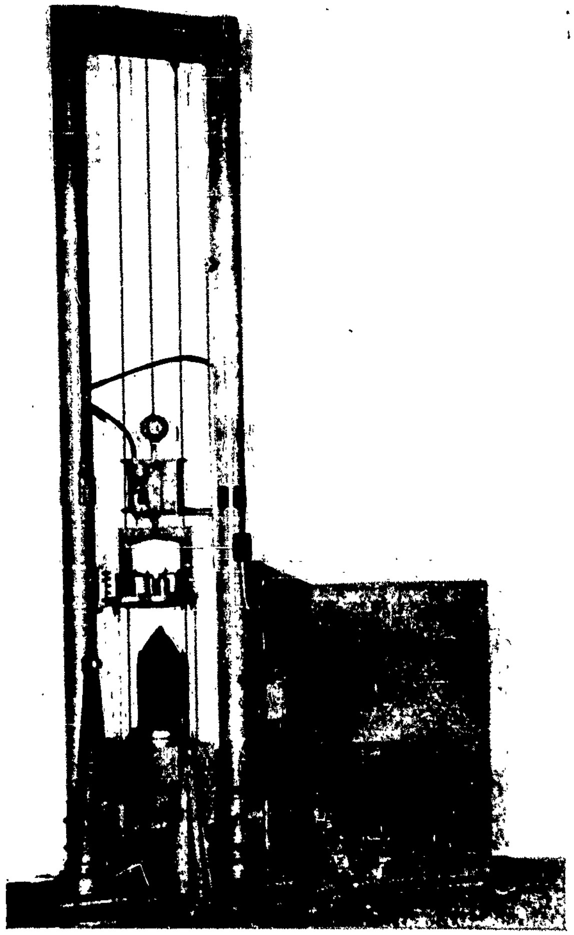
In the array of exhibits at the Symposium was that of the Naval Ordnance Laboratory which featured a 100-ft Drop Tester (see Figure 1, p. 112). The display was arranged by V. F. deVost, Shock and Vibration Branch, Mechanics Division, NOL.

The acceleration-time oscillogram pictured here was made during the demonstration, to show the reproducible characteristics of the stopping device. The two top records are traces of single drops made at 75 fps on the low-g stopping device. The bottom record represents three superimposed traces of three similar drops made on the same bag.

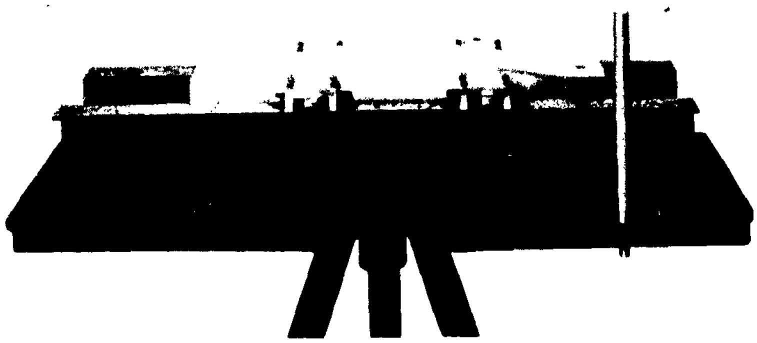
The equipment used to make the traces consisted of a 500-g, 100-cps Statham accelerometer, a type 304H Dumont oscilloscope, and a Polaroid Land camera. The duration time (right to left) is 25 ms and the peak acceleration is 160 g's.



Acceleration time oscillogram



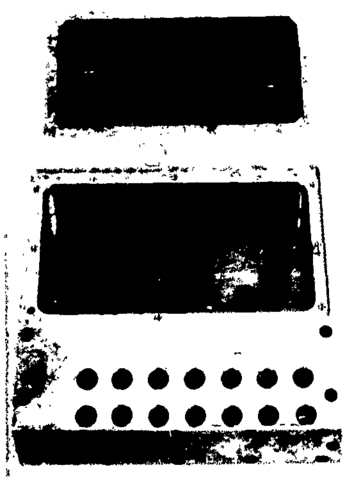
NOL exhibit



Variable cross-section vibrating bar

VARIABLE CROSS-SECTION VIBRATING BAR

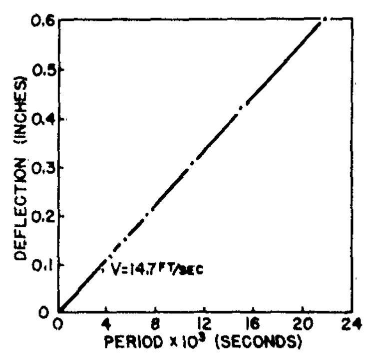
The equipment displayed by the Naval Research Laboratory is being used currently at the Laboratory for shock and vibration work. The exhibit was arranged and conducted by R. W. Conrad, R. J. Peters, and E. W. Clements of the Shock Instrumentation and Measurement Group, Shock and Vibration Branch, Mechanics Division, NRL. Included was a variable cross-section vibrating bar. The design changes may be observed by comparing the photograph of this equipment with the one below, of the uniform cross-section bar. The bar was designed to eliminate the principal causes of failure of present vibrating bars. The accelerometer and armature mounting holes have been moved out to opposite ends at points of zero bending. The center section, now free of stress risers, has been made heavier and the nodes presumably have moved toward the center. As a result, there is less bending in the center, and the lever arm for the accelerometer is longer. The resonant frequency is approximately 1500 cps. This bar has not yet been tested.



D.L.F. gage (high-frequency model)

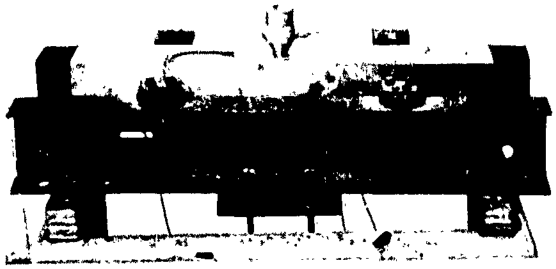
D.L.F.* GAGE (HIGH-FREQUENCY MODEL)

The gage shown here is useful for measuring the damaging effect of shock motions that consist of damped transient disturbances. In providing an analysis, the gage does not indicate what the shock motion is; rather it indicates what the shock motion does. Damping is made tolerable by having no backing for the foil in back of the scribes. The unit illustrated is intended for use in equipment where the component parts have natural frequencies above 40 cps. The reed frequencies are 46, 67, 92, 139, 84, 231, and 278 cps. Displacements can be read to within several thousandths of an inch.



Response of D.L.F. gage to a sudden velocity change of 14.7

*D.L.F. refers to Dynamic Load Factor. This gage is commonly known as the reed gage.



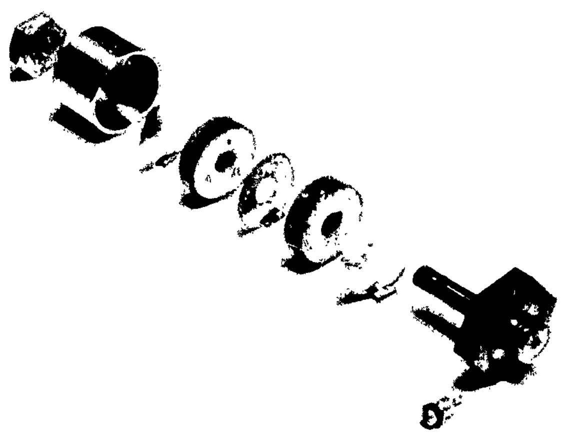
Vibrating bar accelerometer calibrator

VIBRATING BAR ACCELEROMETER CALIBRATOR

This steel bar is 21 in. long with a uniform rectangular cross section and is supported at the nodes by piano wire. The accelerometer is mounted at the exact center of the bar. Directly below the accelerometer is attached an armature which receives its driving power from a 500-watt source. The resonant frequency of the bar is 900 cps and its excursion is measured by a traveling microscope or strain gage. The normal acceleration range for extended periods is about 800 g's. For accelerations as large as 2000 g's, the bar cracks from fatigue in about five minutes.

BARIUM TITANATE ACCELEROMETER, NRL TYPE C-4

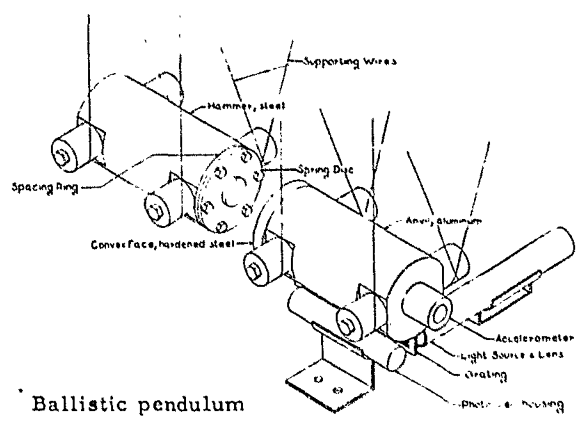
The mounted height of the C-4 is slightly over one inch and it weighs approximately 5 oz. A long bell-shaped brass weight extending completely over the barium titanate elements as a protective cover, gives the accelerometer a lower center of gravity than if the same mass were concentrated in the form of a solid cylinder. With this weight, the transverse sensitivity is approximately 10 percent. Two barium titanate disks slightly under one inch in diameter are positioned back-to-back mechanically, thus putting them electrically in parallel. A hex nut tightened with a torque of about 120 in.-lb provides sufficient tension in the central shaft to keep the elements from becoming unseated for large negative accelerations. For a range of 60 g's to over 7000 g's, the output of the accelerometer is linear and has a sensitivity of 2.10×10^{-10} coulombs per g (157 mv per g, open circuit).



Barium titanate accelerometer, NRL type C-4

BALLISTIC PENDULUM

The length of the ballistic pendulum is 104 in. and its period is 3.26 sec. The accelerometer to be tested is mounted on the forward anvil face, with its lead supported so as not to interfere with the free movement of the anvil. For testing, the hammer is pulled back to a predetermined height and released by an electromagnet. Since the anvil is at rest prior to the blow, it is necessary only to measure its velocity immediately after the blow. Several methods are available for doing this, namely: (1) photoelectric method; (2) determination of the time required to traverse a known distance; and (3) measurement of the height to which the anvil rises after the blow. Velocity changes up to about 10 fps are possible. The duration of the acceleration pulse can be varied from 0.3 ms for a steel-on-steel impact to about one ms for a "soft" blow with a flexible nose piece. Ductile materials, such as lead, interposed between the striking surfaces permit further lengthening of the pulse to 4 or 5 ms. Peak accelerations up to 2500 g's can be produced by this arrangement.



Ballistic pendulum

TELEMETERING EQUIPMENT

A section of the NRL exhibit showed progress made at the Laboratory in telemetering techniques which provide shock data. The display was arranged by E. E. Bissell; Field Engineering Group, Structures Branch, Mechanics Division, NRL.

The equipment exhibited was constructed in order to record signals from instruments located in a submerged target, subject to large underwater explosions. Telemetering the signals from the target to remotely-located recording instruments, is considered essential. Studies made on the equipment indicate that in order for the results to be satisfactory, certain characteristics are essential:

1. Frequency response should be flat from dc to at least 20 kc.

2. FM should be used since it will give the best radio transmission characteristics.

3. The system must be remotely operated via an independent radio-command circuit.

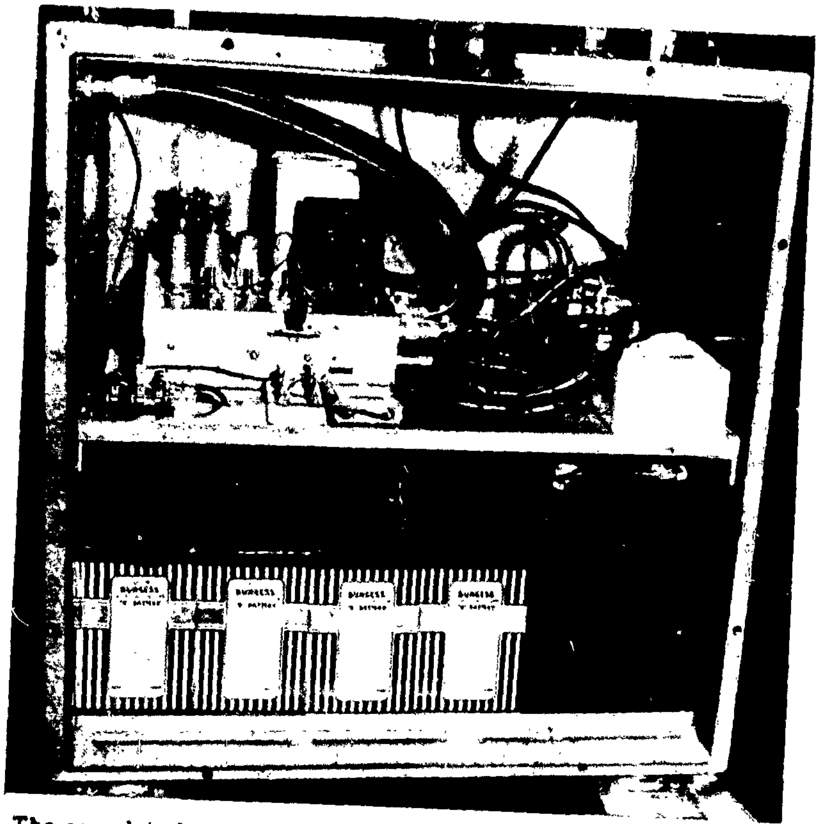
4. An accurate means of calibration prior to the shock is necessary.

5. The system should be capable of operation within optical line-of-sight distances.

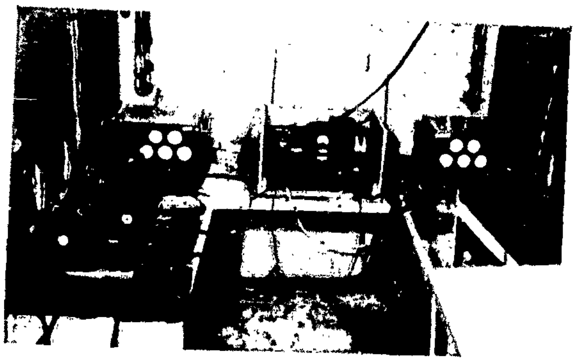
Since no commercial system was available having all these characteristics, it was necessary for NRL to develop a radio command system and combine it with available instrumentation. The command system provides remote-controlled, on-or-off operation of the FM telemetering transmitters. This system was made of an AM transmitter, an AM receiver, and a sequence relay system. The FM telemetering transmitters and the AM command receiver, together with the sequence relay system and necessary power, usually are remotely located on the buoy over the target some distance from the recording instrumentation and the AM command transmitter.

In operation it is necessary that the AM command receiver be turned on manually a few hours prior to the explosion. Ten minutes before the explosion occurs, the FM telemetering transmitters are remotely energized and calibrated by transmitting a 400-cycle audio signal via the AM command transmitter. This signal, received by the AM command receiver located on the buoy, is demodulated, amplified, and limited. The demodulated signal is used to actuate the sequence relay system which turns on the power and provides the amplitude calibration signal which is then retransmitted via the FM telemetering transmitters back to the recording instrumentation. Then when the gain levels on the recording instruments are adjusted, the 400-cycle signal is turned off. Five minutes before the explosion occurs the 400-cycle signal is again turned on; this actuates the sequence relay system which removes the calibration signal from the FM telemetering transmitters. The system is now ready for the explosion. After the explosion, the 400-cycle signal is again turned on and the sequence relay system removes all power from the FM telemetering transmitters. If necessary, this sequence may be repeated.

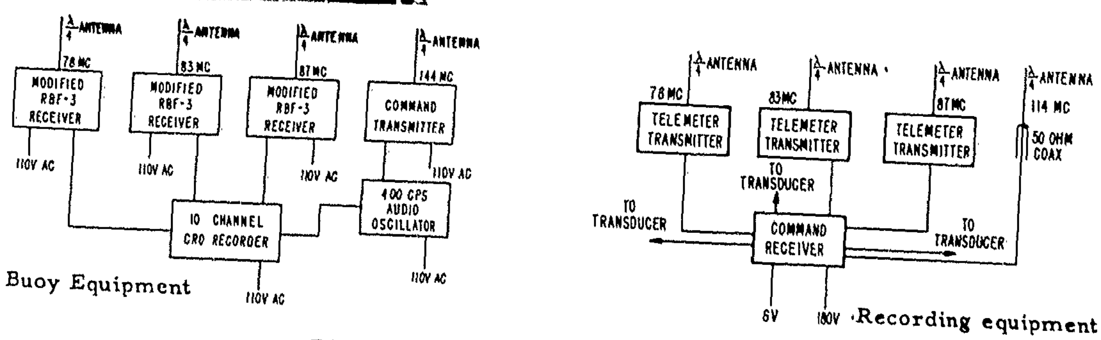
The system was employed in the Caribbean Sea south of Vieques Island in conjunction with tests conducted by the Underwater Explosion Research Division of the Norfolk Naval Shipyard. Three telemetering channels were used to transmit signals from transducers located on the target. The system so far has given good results.



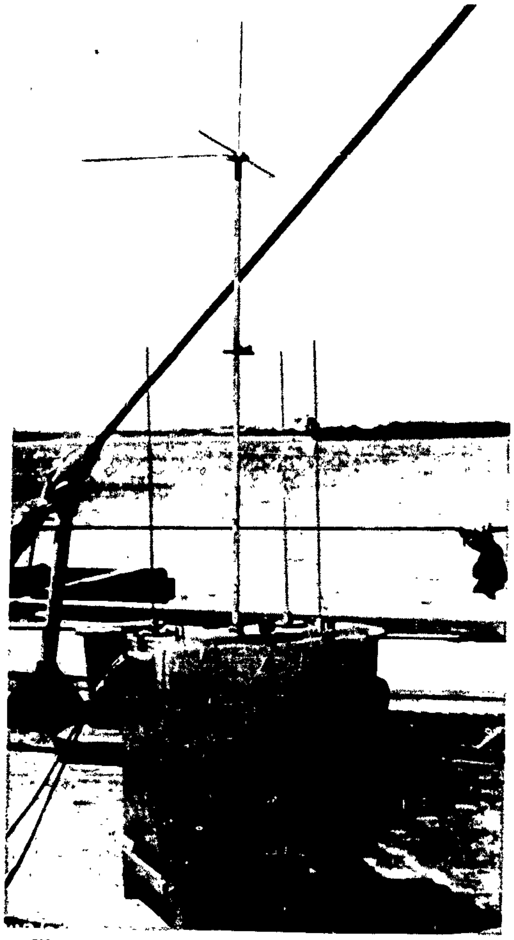
The complete buoy equipment: FM telemetering transmitter, AM command receiver, sequence relay system, and power supply



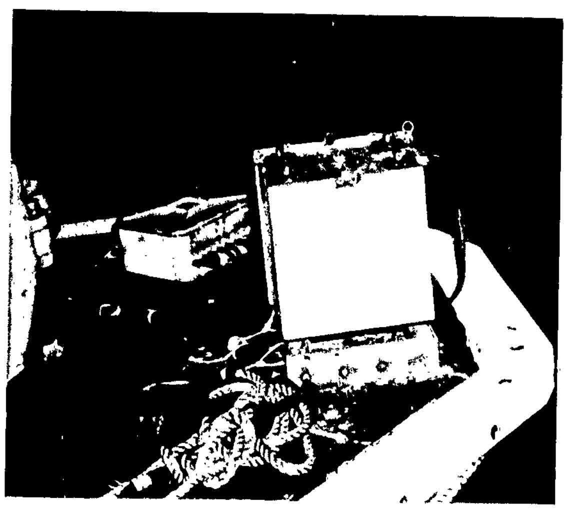
Recording instruments, housed in a van-type trailer which may be loaded on a naval vessel



Block diagram of telemetering system

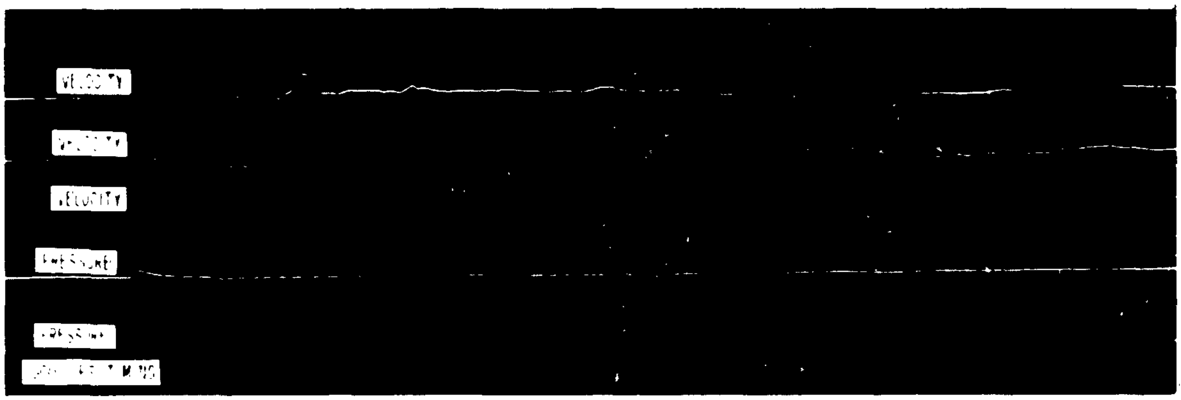


Water-tight housing for buoy equipment

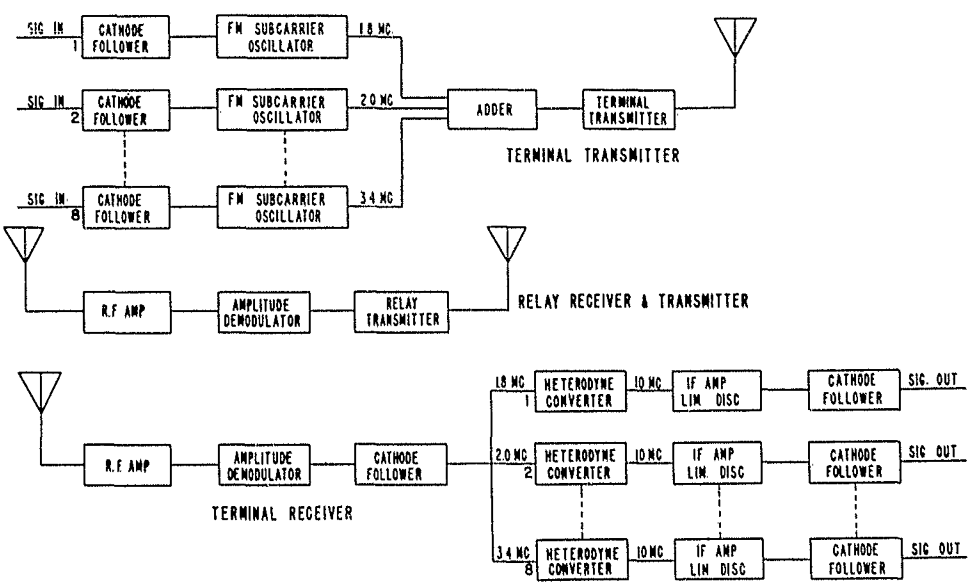


Buoy equipment mounted on a large
telephone-type buoy

Buoy equipment at sea mounted on a large
cylindrical buoy



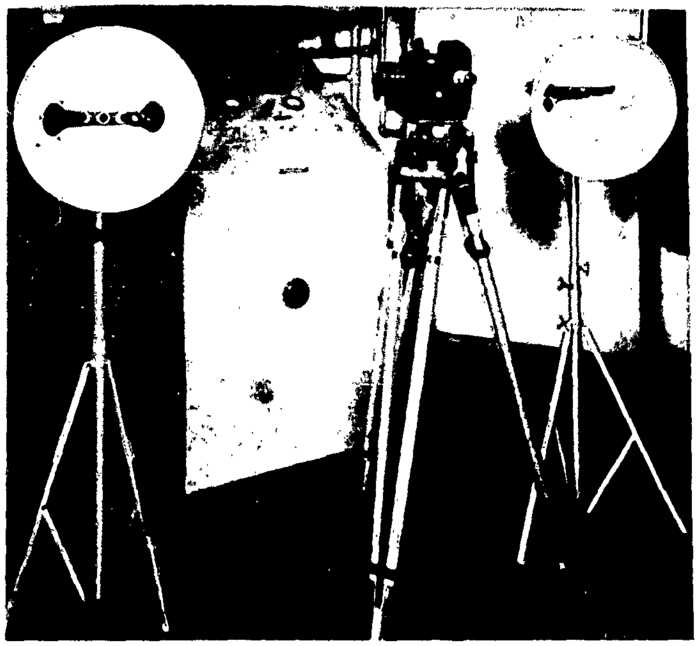
A typical telemeter record



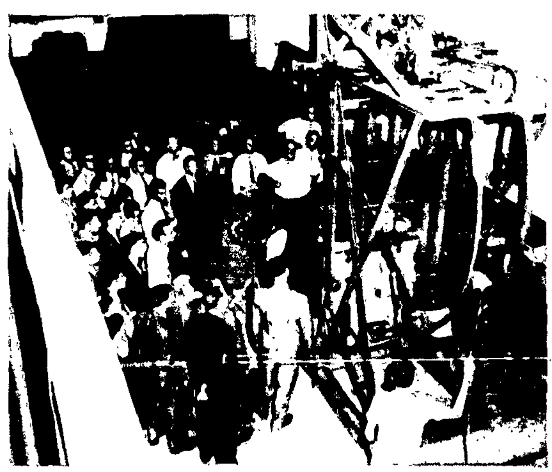
Block diagram of a new telemetering system being developed by NRL. This system will be capable of transmitting 8 channels of information with a frequency response from dc to 10 kc.

NRL EXHIBIT

The Test and Evaluation Group, Shock and Vibration Branch, Mechanics Division, NRL, set up a display of the use of a high-speed movie camera with the Navy light-weight, high-impact shock machine. The high-speed movie camera was also put into the NRL exhibit as an example of equipment in use at the Laboratory. C. C. Hauver, Photographic Technologist, Mechanics Division, NRL, was in charge of this part of the NRL exhibit.



High-speed movie camera



High-speed movie camera used in demonstration with Navy light-weight, high-impact shock machine